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EVALUATION OF LAMINATED ELASTOMERIC BEARINGS IN THE UH-1 HELICOPTER TAIL ROTOR

By

C. H. ~~Fagen~~

July 1967

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

CONTRACT DA 44-177-AMC-313(T)

BELL HELICOPTER COMPANY
A DIVISION OF BELL AEROSPACE CORPORATION
FORT WORTH, TEXAS

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Laminated elastomeric bearings offer the potential of eliminating lubrication, reducing maintenance, and increasing the service life of helicopter tail rotor systems. The objective of this program was to evaluate the use of laminated elastomeric bearings in a helicopter tail rotor system.

The report describes the design, fabrication, and flight tests of the bearings. Results of the tests indicate that the potential advantages are obtainable.

This command concurs in the conclusions contained herein.

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EVALUATION OF LAMINATED ELASTOMERIC BEARINGS IN THE UH-1 HELICOPTER TAIL ROTOR

BHC Report 572-099-004

By

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SUMMARY

This report presents the results of a research program conducted to establish a sound basis for evaluation of elastomeric bearings and to investigate the bearings' characteristics in a UH-1 helicopter tail rotor assembly. Radial and thrust elastomeric bearings were evaluated in the tail rotor flapping and pitch change axes. The thrust bearings were used to accommodate the blade pitch change motions and to carry the blade centrifugal forces. The radial bearings were used to allow the tail rotor flapping motions and to carry the thrust and drive loads.

Radial and thrust bearings, of the bonded-thin-elastomer-layer type, were bench and whirl tested in the experimental tail rotor. Also, analyses were conducted to establish the tail rotor's airworthiness and dynamic characteristics.

Two elastomeric bearing tail rotor configurations were flight tested. Both used standard UH-1 blades. The first configuration used bonded-thin-elastomer-layer-type bearings in both the pitch change and flapping axes. The second configuration used molded-type elastomeric bearings in both axes. This fabrication process requires thicker elastomer layers. Also, conical steel sheets (120° included angle) separating the elastomer are required in the thrust bearing to provide column stability.

The molded-type bearing was found to be the more promising of the two investigated. Three additional tail rotor configurations, using the molded-type bearings, were flight tested under a Bell Helicopter Company Independent Research Program. Data for all the configurations tested are included herein and are compared with standard UH-1 tail rotor data. The comparisons show that the elastomeric bearing concept offers considerable promise. The principal problem encountered involved achieving satisfactory rotor blade frequency placement to assure acceptable load levels; however, this is not a problem unique to the elastomeric bearing concept. The load path through any bearing arrangement is difficult to evaluate. With the final configuration evaluated, acceptable characteristics were demonstrated. The data from this program will allow a more accurate evaluation of future elastomeric bearing applications.

FOREWORD

This report is submitted in compliance with provisions of USAAVLABS Contract Number DA 44-177-AMC-313(T), "Evaluation of Laminated Elastomeric Bearings in the UH-1 Helicopter Tail Rotor". The work conducted under this program commenced upon receipt of the contract on 24 June 1965.

Bearing design, fabrication, and bench and whirl tests were conducted under phase I of the program. Flight tests of an elastomeric bearing tail rotor were conducted under phase II. The program as originally contracted was completed on 20 June 1966. A second tail rotor configuration was flight tested under Modification 3 to the contract, and these tests were completed on 15 December 1966.

The program was conducted under the technical cognizance of Mr. E. R. Givens of the Aircraft Systems and Equipment Division of USAAVLABS. Principal Bell Helicopter Company personnel associated with the program were Messrs. S. Aker, R. W. Balke, C. R. Cox, W. L. Cresap, C. H. Fagan, R. Lynn, and L. Spencer. Also, the writer wishes to acknowledge the technical assistance of Mr. R. Peterson of Lord Manufacturing Company.

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INTRODUCTION

Laboratory investigations of elastomeric bearings were conducted by Bell Helicopter Company during 1964 and 1965. (Earlier investigations of the bearing concept are reported in Reference 1.) These investigations, conducted under a company Independent Research Program, were concerned principally with exploring bearing fabrication techniques and determining bearing basic characteristics. The results of this work, reported in part in Reference 2, indicated that the elastomeric bearing would offer several advantages over conventional bearings where oscillatory motions are to be accommodated. The elastomeric bearing appeared to be particularly suitable for use in the helicopter tail rotor flapping and blade feathering axes, and further investigations were directed towards the UH-1 tail rotor as a specific application. Analyses and design studies of possible elastomeric bearing tail rotor configurations were conducted, and it was indicated that this type of bearing would be suitable for this application. The potential advantages expected for the elastomeric bearing tail rotor included a reduction of rotor weight, the elimination of lubrication requirements, and an increase in service life.

The expected problems associated with elastomeric bearing fabrication were those of bonding and control of the elastomer quality as well as the layer thicknesses. Also, the durability and spring rates of the elastomeric bearings were expected to be an important factor in the tail rotor operation.

Under the subject contract, Bell Helicopter Company began a program to investigate elastomeric bearings further. The program consists of bearing design, fabrication, and bench test; also, whirl and flight test of an experimental helicopter tail rotor using bonded-type elastomeric radial bearings in the flapping axis and elastomeric thrust bearing to accommodate the blade pitch change motions. The contract was later modified to include testing of a new, recently developed type of elastomeric thrust bearing. This configuration required one bearing in each grip to carry the blade centrifugal force and to accommodate the pitch change motions. New elastomeric radial bearings were also used in the flapping axes for this configuration.

The results of the work performed under this contract are reported herein. Additionally, salient results of the Bell Independent Research work are presented.

BEARING DESIGN AND FABRICATION

Two basic bearing constructions were investigated during this program. The initial configuration consisted of a bonded construction with alternate thin laminates of elastomer and steel shims. Later in the program, a molded-type construction with formed sheets separating the elastomer was investigated. The latter construction was found to be the most promising of the two; consequently, this bearing was the principal type investigated during the latter part of the program. The two bearing constructions are discussed in the following paragraphs. A general description of the bearings investigated is given in Table I (page 3). The bearing descriptions (i.e., "BR1", "BT1", "BT2" bearings, etc.) as given in Table I are also used within this report.

GENERAL DESIGN CONSIDERATIONS

The radial bearing was designed to replace the flapping hinge needle bearing in the standard UH-1 tail rotor. Dimensions of the hub and trunnion established the inside and outside diameter limits.

The thrust bearing was designed to replace the existing tail rotor blade pitch change bearings with one thrust bearing per blade grip. The inside diameter was established by the outside diameter of the rotor yoke spindle, and the outside diameter of the bearing was determined by the requirement that the thrust capacity of the bearing be 16,000 pounds, which is the blade centrifugal force developed at operating speed. Also, the torsional spring rate of the bearing was limited to 20 inch-pounds per degree of rotation per blade to prevent excessive control forces.

BONDED BEARINGS

The material selected for shims in both the thrust and radial thin-layer-type bearings was 301 stainless steel, chemically cleaned and surface-treated with Bonderite 32. Several elastomers were investigated, and silicone XE508 elastomer was selected for this application because of its good bond adhesion qualities.

All of the bonded-type thrust and radial bearings were fabricated by initially "squeezing" the paste-like, uncured silicone elastomer onto surface-treated shims, stacking the elastomer-coated shims on pins of a plate tool (see Figures 1 and 2), and compressing the resultant stack to give the desired elastomer layer thickness.

TABLE I. DESCRIPTION OF ELASTOMERIC BEARINGS

Bearing Configuration	Bearing Description	Bearing Tests and Results
BR1 Bonded Radial	Radial Bearing (BHC Part No.572-018-003-1) consists of 31 layers of elastomer, .004-inch thick, separated by .002-inch thick shims. Bearing envelope dimensions are O.D. 1.375 inch, I.D. .625 inch, and length 1.00 inch.	Bench tests showed the bearing torsional spring rate to be acceptable for the tail rotor application. One bearing failed after 178 hours of endurance testing, and another failed after 380 hours of the same type of test. Two bearings were in good condition after 30 hours of whirl testing. *
BT1 Bonded Thrust	Thrust Bearing (BHC Part No.572-018-004-1) consists of 75 layers of elastomer, .004-inch thick, separated by .002-inch thick steel shims. Bearing envelope dimensions are O.D. 2.68 inch, I.D. 1.75 inch, and length .648 inch.	Two bearings were permanently distorted due to column buckling at loads above 10,000 pounds. Other bearings were tested, and torsional and compression spring rates were determined. Two bearings extruded elastomer during whirl test.
BT2 Bonded Thrust	Thrust Bearing (BHC Part No.572-018-004-3) consists of 75 layers of elastomer, .0034-inch thick, separated by .002-inch thick steel shims. Bearing dimensions are O.D. 2.68 inch, I.D. 1.75 inch, and length .648 inch.	Two bearings extruded some elastomer during whirl test (installed in outboard location of grips). Failure was caused by unequal distribution of the centrifugal force. The other two bearings were in good condition after whirl test 1 (installed in inboard location of grips). Other bearings were in good condition after bench, whirl, and ground and flight tests on the helicopter.

*For the radial bearing endurance test, 1 hour is equal to 1-hour bearing service life at 120 knots. The radial bearings do not oscillate during whirl test.

TABLE I. - Continued

Bearing Configuration	Bearing Description	Bearing Tests and Results
MT1 Molded Thrust	Thrust Bearing (Lord Mfg. Co. Part No. J12824-1) consists of 11 layers of 56-durometer polymer-blend compound, .020-inch thick, separated by .020-inch thick, 10-degree formed steel sheets. Bearing envelope dimensions are O.D. 2.680 inch, I.D. 1.750 inch, and length .648 inch.	Bearing was in good condition after whirl test and helicopter ground run.
MR1 Molded Radial	Radial Bearing (Lord Mfg. Co. Part No. J12727-1) consists of 5 concentric layers of 56-durometer polymer-blend compound, .025-inch thick, separated by concentric cylinders which have wall thicknesses of .025 inch. Bearing envelope dimensions are O.D. 1.375 inch, I.D. .625 inch, and length 1.00 inch.	One bearing was in good condition after 560 hours of endurance testing. It was bench tested at -60°F and found to have a spring rate 7.9 times that at room temperature. Other bearings were in good condition after pre-flight whirl test and helicopter ground and flight tests.
MT2 Molded Thrust	Thrust Bearing (Lord Mfg. Co. Part No. J12824-6) consists of 27 layers of BTR-4 elastomer, .025-inch thick, separated by .025-inch thick, 30-degree formed steel sheets. Bearing envelope dimensions are O.D. 2.680 inch, I.D. 1.75 inch, and length 1.75 inch.	Two bearings were in good condition after pre-flight whirl test, helicopter ground run, and flight tests. Another bearing was bench tested to determine bearing torsional and compression spring rates.

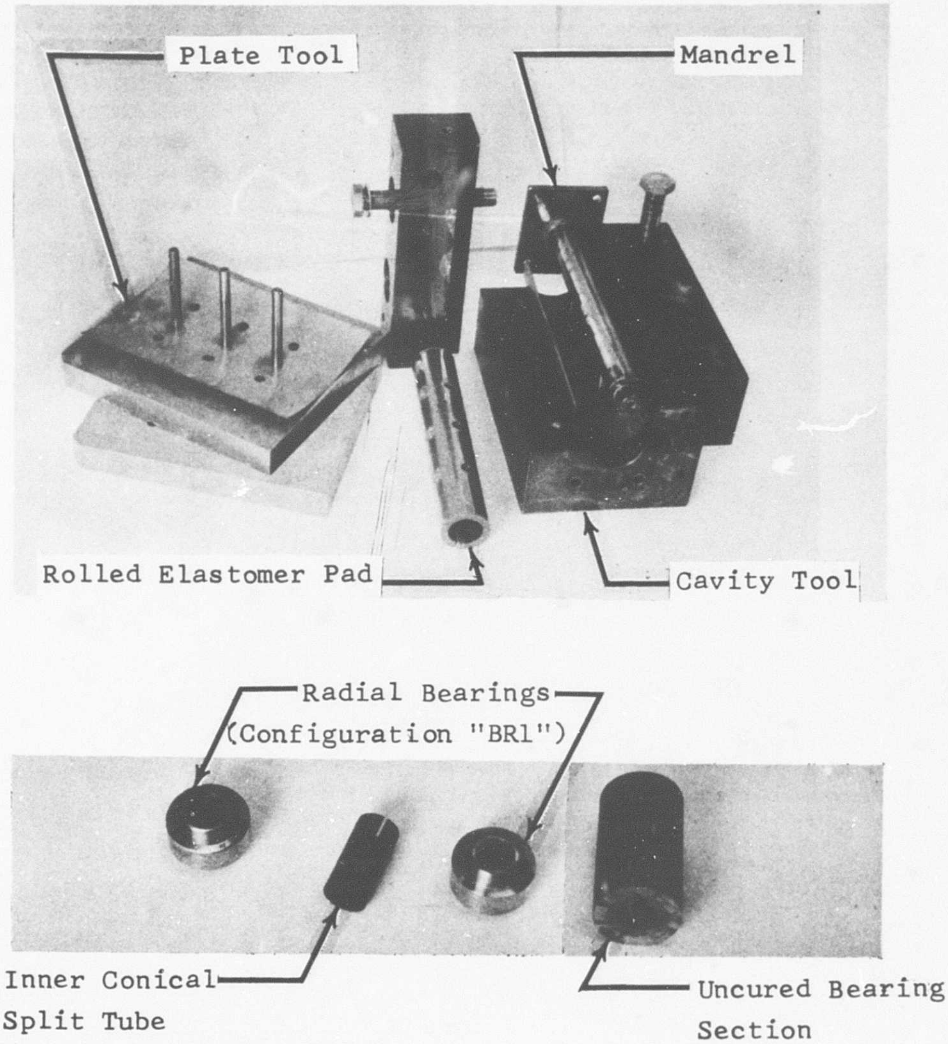


FIGURE 1. FABRICATION PROCESS FOR BONDED-
THIN-ELASTOMER-LAYER RADIAL BEARING.

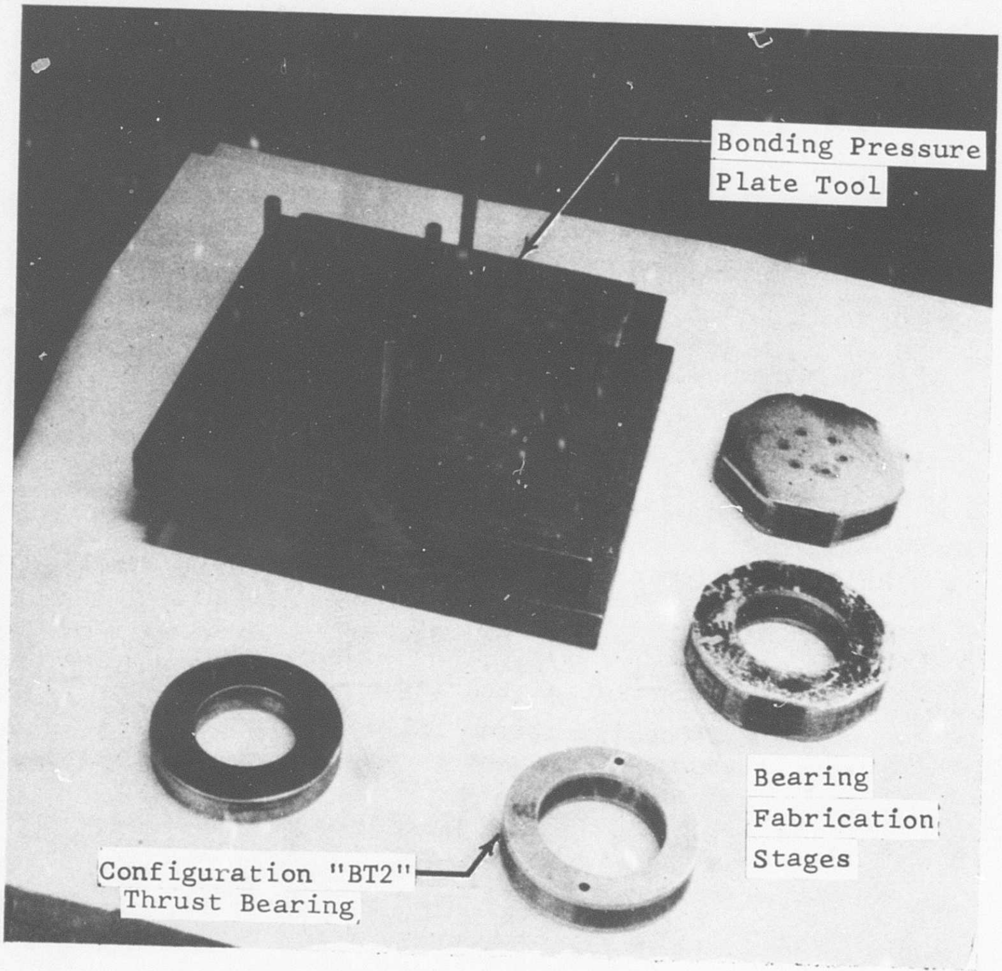


FIGURE 2. FABRICATION PROCESS FOR BONDED-THIN-ELASTOMER-LAYER THRUST BEARING.

Radial Bearings

The radial bearing was tapered in width from the inside laminate to the outer laminate, such that the calculated shear strain factor was equal in all layers of elastomer for torsional deflections. Radial bearings, Configuration "BR1" in Table I, were designed with 31 layers of .004-inch-thick elastomer. The calculated shear strain factor of .39 for 5-degree rotation was considered to be satisfactory for this oscillatory application (shear strain factor is defined as the displacement between the two surfaces of the laminate divided by its thickness).

During the manufacturing process, the radial bearing stock slab is removed from the plate tool immediately after the slab is formed to thickness. It is then wrapped around a mandrel and inserted into a cylindrical cavity where it is cured to form a cylindrical unit (see Figure 1). The unit is next cut into four segments, and each segment is coated with elastomer and inserted into a steel tube which, after machining, becomes the radial bearing outer ring. A split tube with a conical inner surface (see Figure 1) is pressed into the laminated segment, and then another tube with a conical outer surface is pressed into the split tube, causing it to expand and compress the laminated segment against the outer ring. The assembly is placed in an oven and cured at 350°F for 24 hours. Bonding pressure is applied by thermal expansion.

After cure, the components are cooled to -100°F, and the inner conical pin is forced further into the unit. This operation is necessary to eliminate the residual tensile stress in the elastomer which results from cooling shrinkage after cure. To finish the bearing, the pin is bored out to the desired diameter, and the outer surfaces are finished.

Thrust Bearings

The slab formed for thrust bearings by the bonding operation is cured on the plate tool. Four bearings are machined from the 6-inch-square cured slab shown in Figure 2.

The original thrust bearing design (Configuration "BT1") consisted of seventy-five .004-inch-thick layers of elastomer, separated by .002-inch steel shims. The calculated shear strain factor for 12-degree rotation was .95, and the column stability was considered sufficient, based on available test data. Subsequent tests proved the column stability of this bearing to be inadequate, and conic-type thrust bearings were investigated as a means of increasing the bearings' column stability; however, the shims proved to be too thin for the forming operation. Therefore, the design was changed to

distribute the 16,000-pound centrifugal load of one blade between two thrust bearings.

The new thrust bearing (Configuration "BT2") was designed consisting of seventy-five layers of .0034-inch thickness, which later proved to be the maximum silicone elastomer thickness that would not extrude under the design load. The calculated shear strain factor of this bearing with 12-degree rotation was 1.11.

MOLDED BEARINGS

The molded-type bearing requires thicker layers of elastomer than the bonded bearings due to the hot mold transfer method of fabrication. Also, thicker separators are required which are placed in the mold prior to the hot elastomer transfer during fabrication.

Radial Bearings

The radial bearings (see Figure 3) consist of five concentric layers of .025-inch-thick, 56-durometer polymer-blend compound, separated by .025-inch-thick concentric cylinders, and are fabricated by the hot transfer mold method. Three bearings (Configuration "MR1") were procured; one was bench tested, and two were tested in the tail rotor during the single-bearing-per-grip configuration flight test program.

Thrust Bearings

Four thrust bearings (Configuration "MT1") were procured for evaluation in the dual-bearing-per-grip configuration. The bearing envelope dimensions were the same as the previous "BT2" bearing; however, the elastomer section consisted of eleven .020-inch-thick layers of 56-durometer polymer-blend compound, separated by 10-degree, .025-inch-thick formed steel plates.

During the latter part of the original flight test program, new elastomers and molded thrust bearings were investigated for use in a single-bearing-per-grip tail rotor configuration. The bearing requirements included increased column stability to carry the 16,000-pound blade centrifugal force as well as a low torsional spring rate of 13 to 17 inch-pounds per degree and not more than three times this value at -60°F. The new thrust bearing design (which met these requirements) consisted of 27 layers of BTR-4 elastomer, .025-inch thick, separated by .025-inch-thick, 30-degree formed sheets. Three bearings (Configuration "MT2") were procured for the bench and flight test program extension. Figure 3 shows the 30-degree steel sheet separators in a section which has been removed from the molded (Configuration "MT2") thrust bearing.

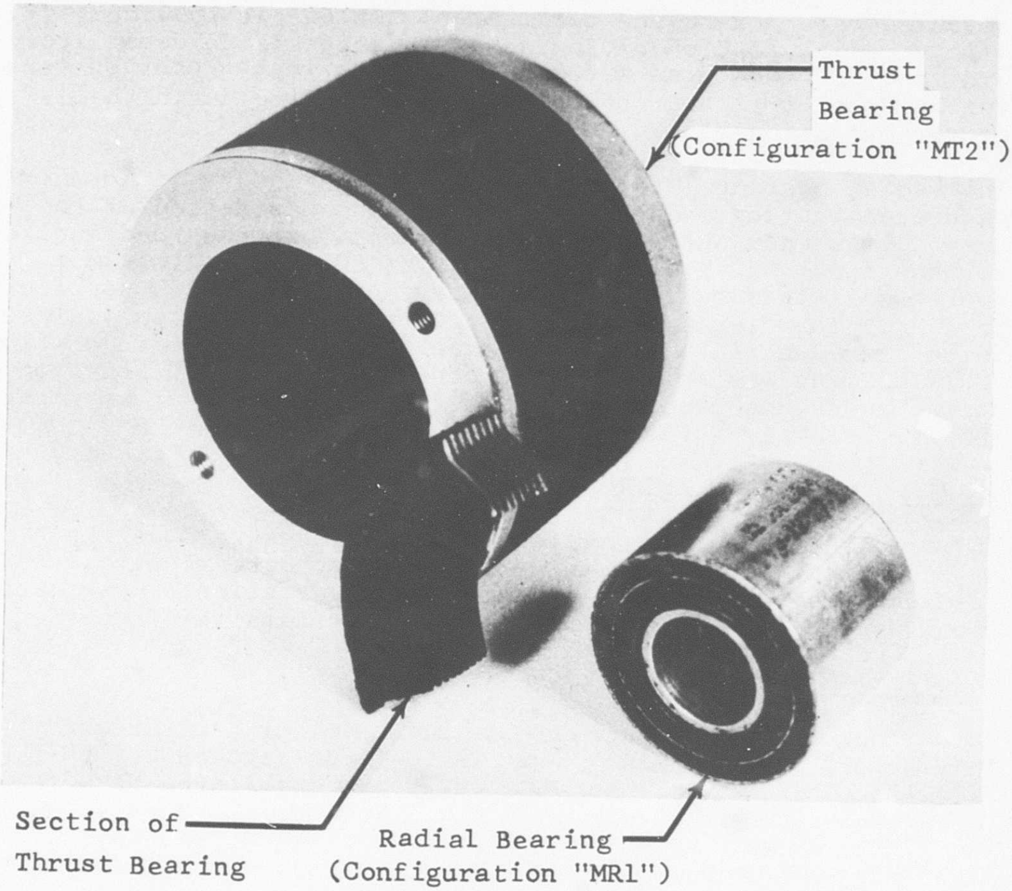


FIGURE 3. MOLDED-TYPE BEARINGS.

BENCH TESTS

ENDURANCE TESTS

The normal operating radial bearing endurance test conditions selected were ± 5-degree oscillatory motion, at 1650 cpm, with a radial load of 570 pounds. These tests of oscillatory and loading conditions were selected such that 1 hour of testing is equal to 1 hour of bearing service life during helicopter operation at 120 knots forward flight.

One radial bearing, "BR1", failed after 178 hours of normal endurance testing. Another "BR1" bearing was tested at the same normal endurance conditions, except that the load applied for the first 150 hours of testing was 800 pounds (maximum overload). This bearing failed after a total of 380 hours of testing. The bearings' temperatures remained constant at room temperature until the latter stages of failure. Configuration "MR1" bearing was still in good condition after 560 hours of normal endurance testing. The sketch in Figure 4 illustrates the test machine for this test.

BEARING DEFLECTION TESTS

The bonded-type thrust and radial bearings as well as the molded radial bearing were bench tested to determine their load-carrying ability and damping characteristics. Test specimens were loaded three times before test data were recorded.

Angular Deflection

Rotational deflection versus torsional moment data are shown in Figure 5 for the "BR1" bearing for temperatures of -60°F, 72°F, and 180°F. Data for the "MR1" bearing, taken at a temperature of -60°F, are shown in Figure 6. Torsional deflection data for thrust bearings "BT1", "BT2", and "MT1", taken at room temperature, are shown in Figure 7. The hysteresis loops shown in Figures 5, 6, and 7 represent the energy loss due to internal friction in the bearings during their torsional deflections.

Compressive Deflection

The thrust bearings were loaded in compression, and their spring rates were determined. The compressive spring rate of "BT2" bearing was found to be 1,500,000 pounds per inch, and that determined for "MT2" bearing was 143,000 pounds per inch.

The bonded radial "BR1" bearing was loaded radially, and the results obtained at temperatures of -60°F, 72°F, and 180°F are shown in Figure 8. The molded radial "MR1" bearing was similarly tested, and the results (at 72°F) are also shown in Figure 8.

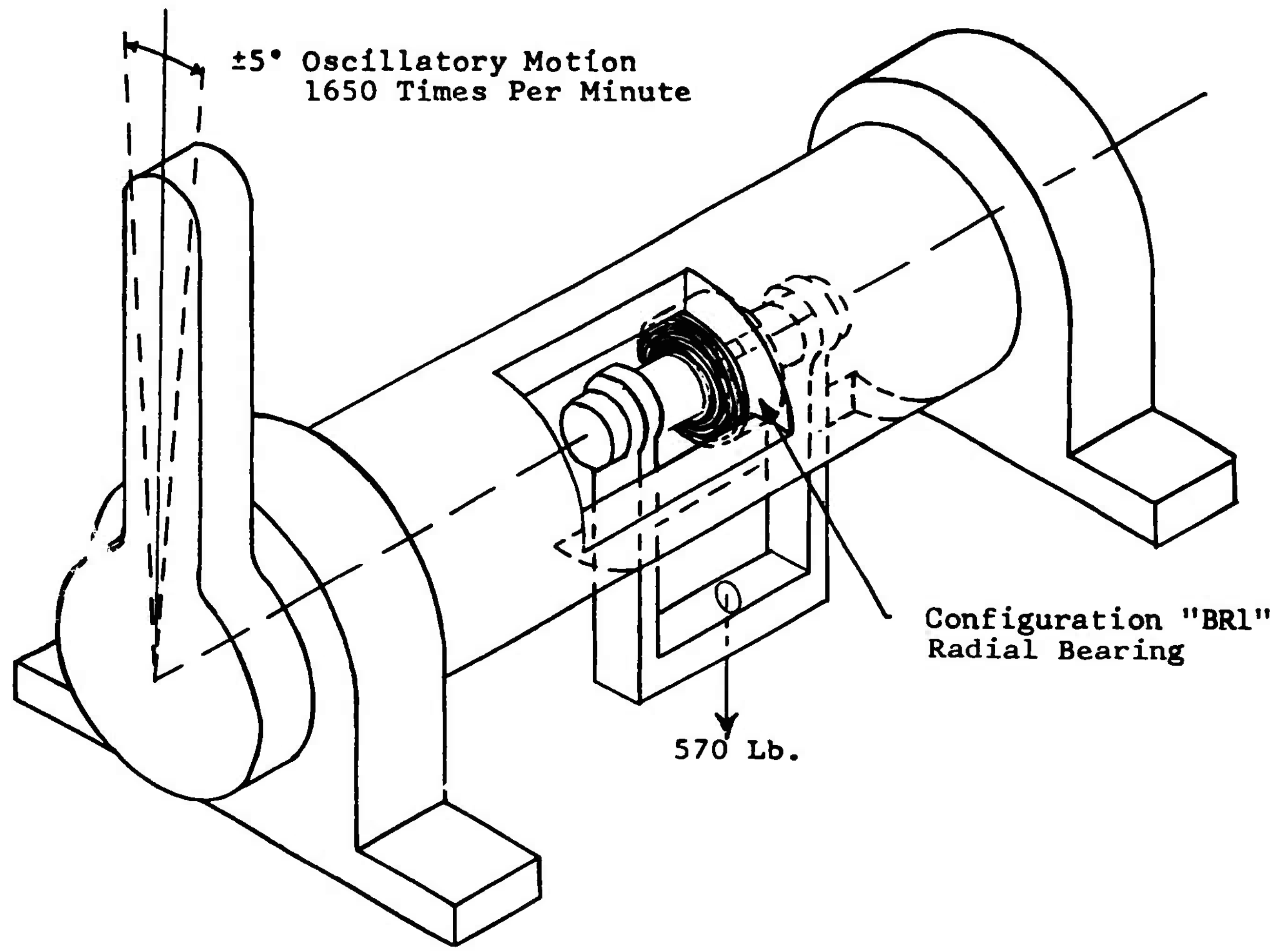


FIGURE 4. SKETCH OF ELASTOMERIC RADIAL BEARING TEST MACHINE

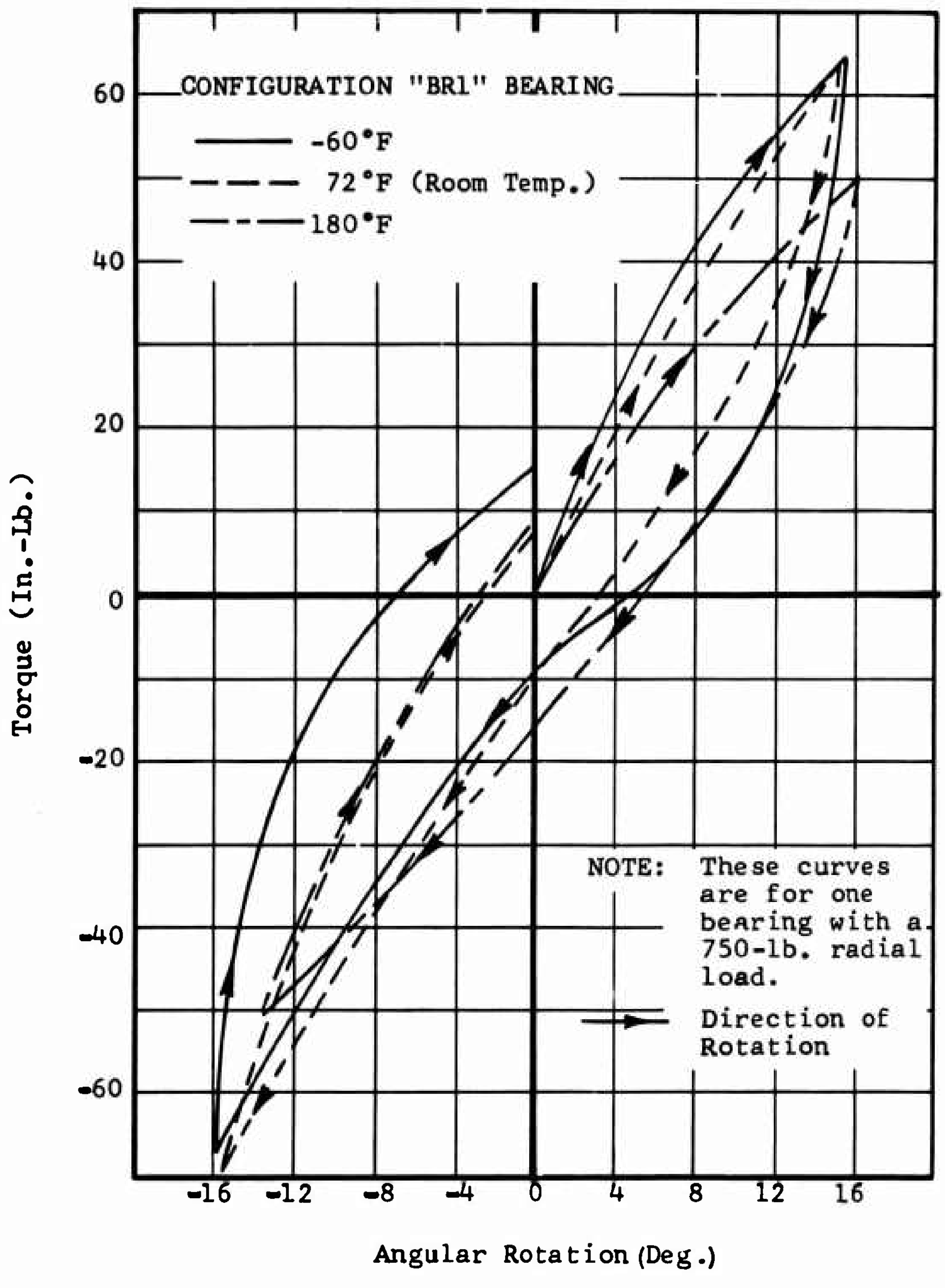


FIGURE 5. ROTATIONAL DEFLECTION VERSUS TORQUE (BONDED RADIAL BEARINGS).

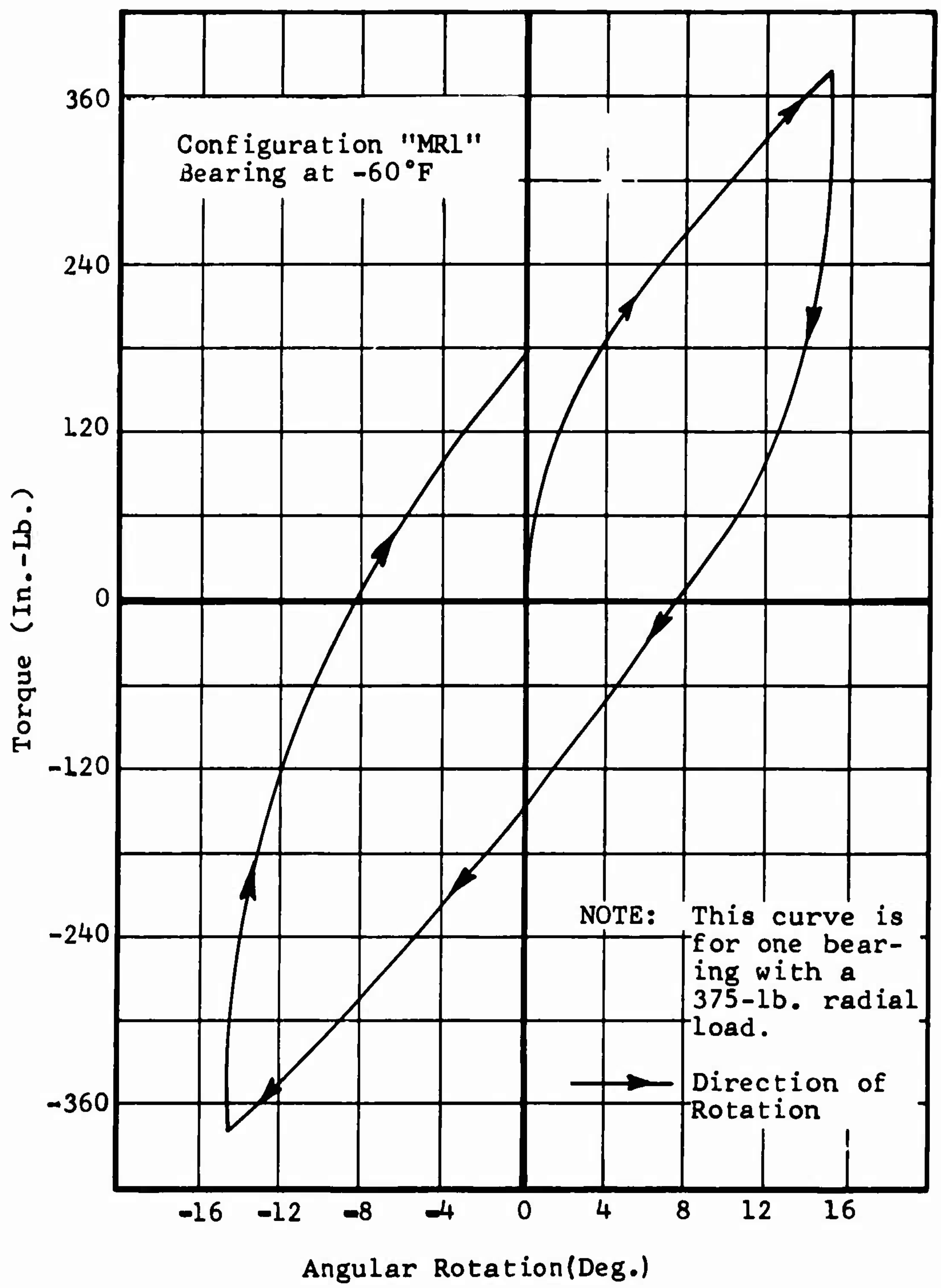


FIGURE 6. ROTATIONAL DEFLECTION VERSUS TORQUE (MOLDED RADIAL BEARINGS).

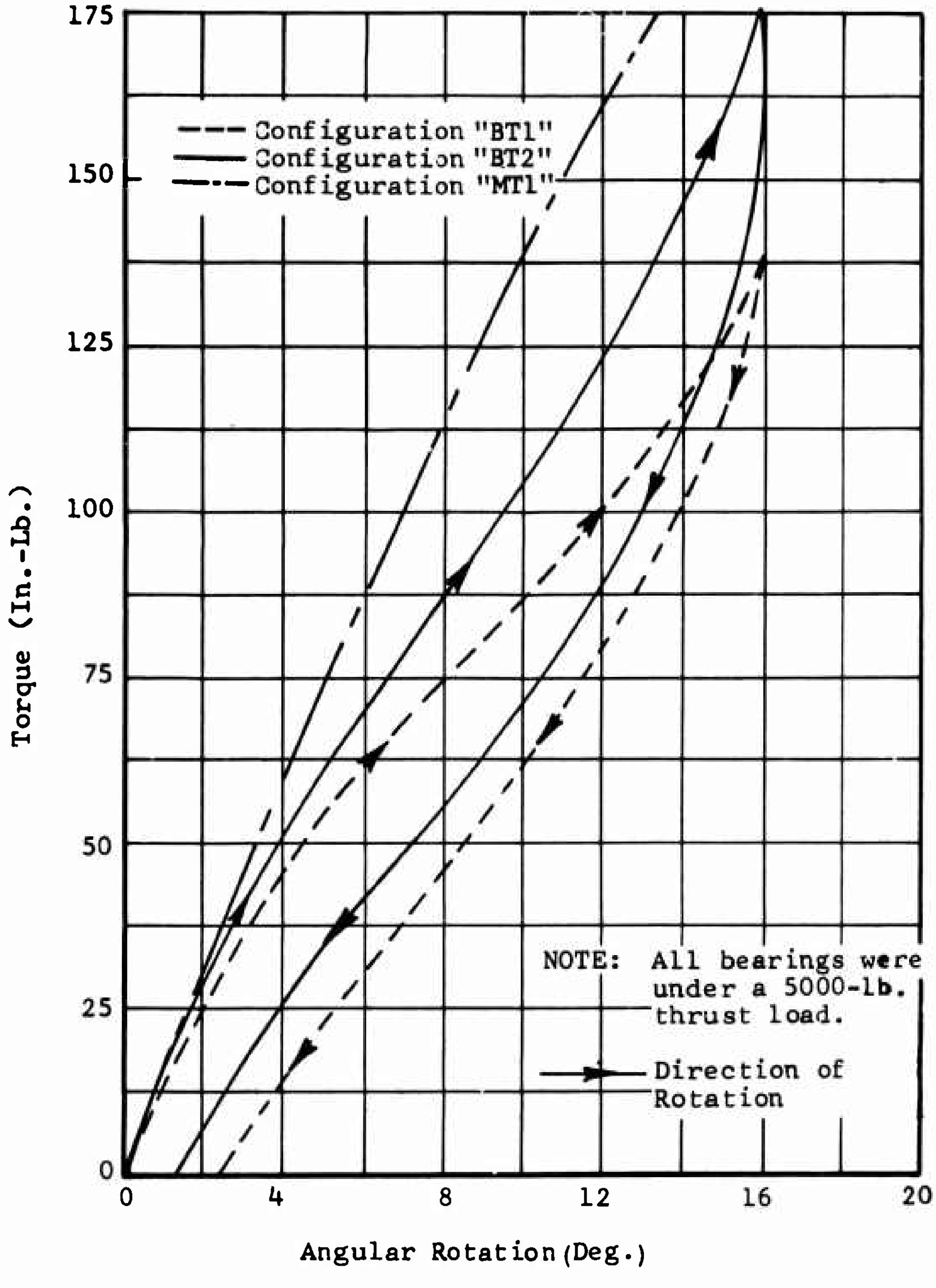


FIGURE 7. ROTATIONAL DEFLECTION VERSUS TORQUE (THRUST BEARINGS).

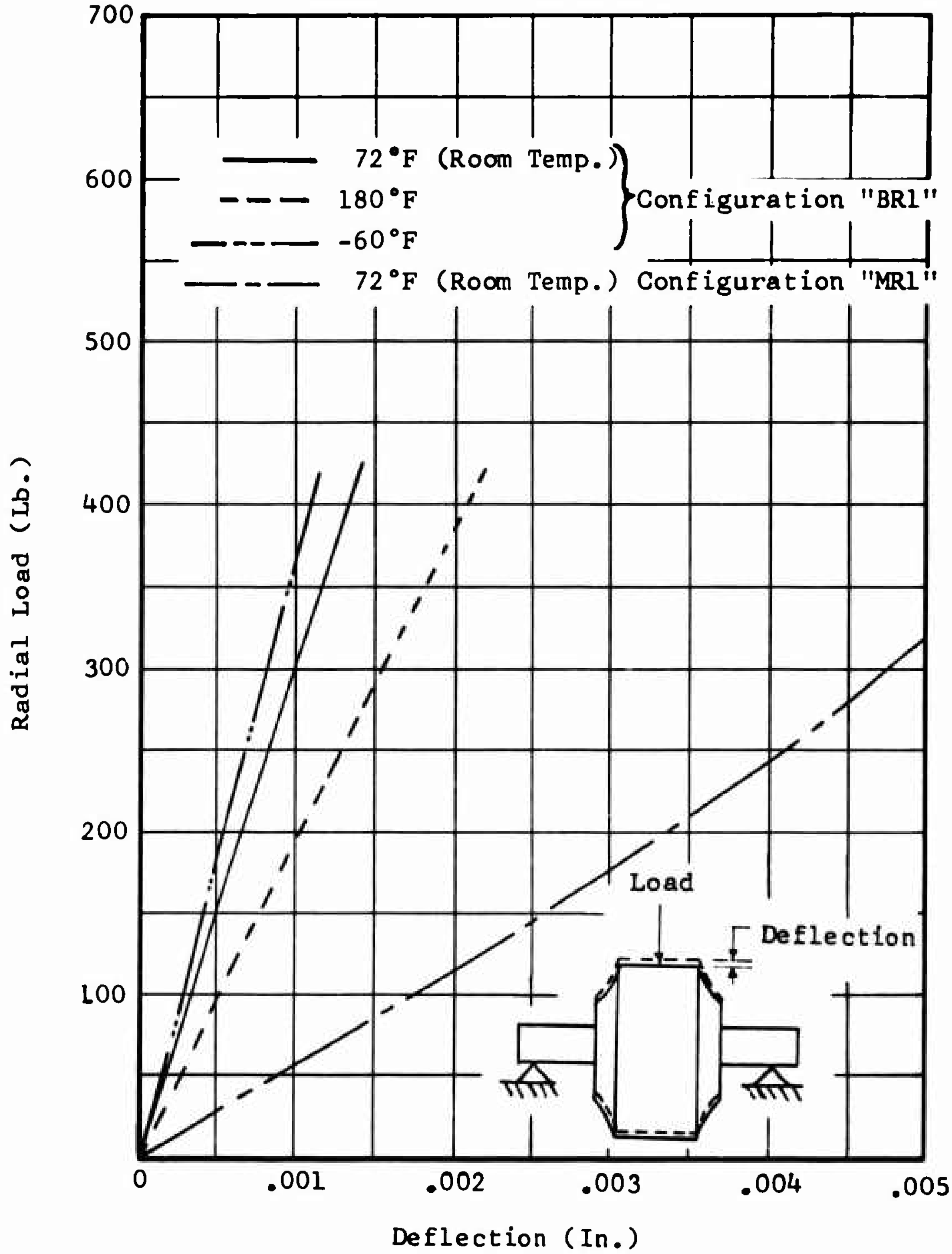


FIGURE 8. RADIAL DEFLECTION VERSUS LOAD (RADIAL BEARINGS).

WHIRL TESTS

Whirl tests were conducted to evaluate the experimental tail rotor with the elastomeric bearings installed. Three separate tests of 10 hours duration each were conducted to obtain an indication of bearing service life and to obtain rotating frequency data. Also, two tests of .5 hour each were conducted for preflight checkout.

TEST PROCEDURE

The whirl tests were conducted using the setup as shown in Figure 9. The test conditions were selected such that 10 hours of whirl test is representative of a UH-1 tail rotor in field service for a period of 1 year (900 hours of flight at four normal rudder reversals per hour). The test rotors were operated at 1650 rpm (equivalent to 324 main rotor rpm), and the blade pitch was cycled between -5 degrees and 12 degrees at 6 cpm.

TEST RESULTS

Four Configuration "BT2" bearings were evaluated during the initial 10-hour whirl test. The bearings were inspected at the conclusion of the test, and it was found that the inboard and outboard bearings were unequally loaded. Figure 10 shows the inboard and outboard bearings removed from one grip. The outboard bearings had extruded elastomer and also showed evidence of rubbing on the housing, indicating that column buckling had occurred. Both inboard bearings were in good condition.

A second 10-hour whirl test was conducted with four Configuration "MT1" bearings. Prior to this test, the hub adjustment method was changed to provide for equal loading of the bearings. At the conclusion of this test, all four bearings were found to be in good condition. The steady pitch link loads with these bearings installed (see Figure 11) indicated that the torsional spring rate was greater than for the previous test configuration and also well above that of the standard UH-1 tail rotor.

A third 10-hour whirl test was conducted with Configuration "BT1" bearings installed in one grip and Configuration "BT2" bearings installed in the opposite grip. At the conclusion of this test, the Configuration "BT1" bearings showed evidence of elastomer extrusion and also column buckling. The Configuration "BT2" bearings were both in good condition. The condition of the Configuration "BT2" bearings of this test as compared with the initial tests (also "BT2" bearings) indicates the importance of equalizing bearing loads.

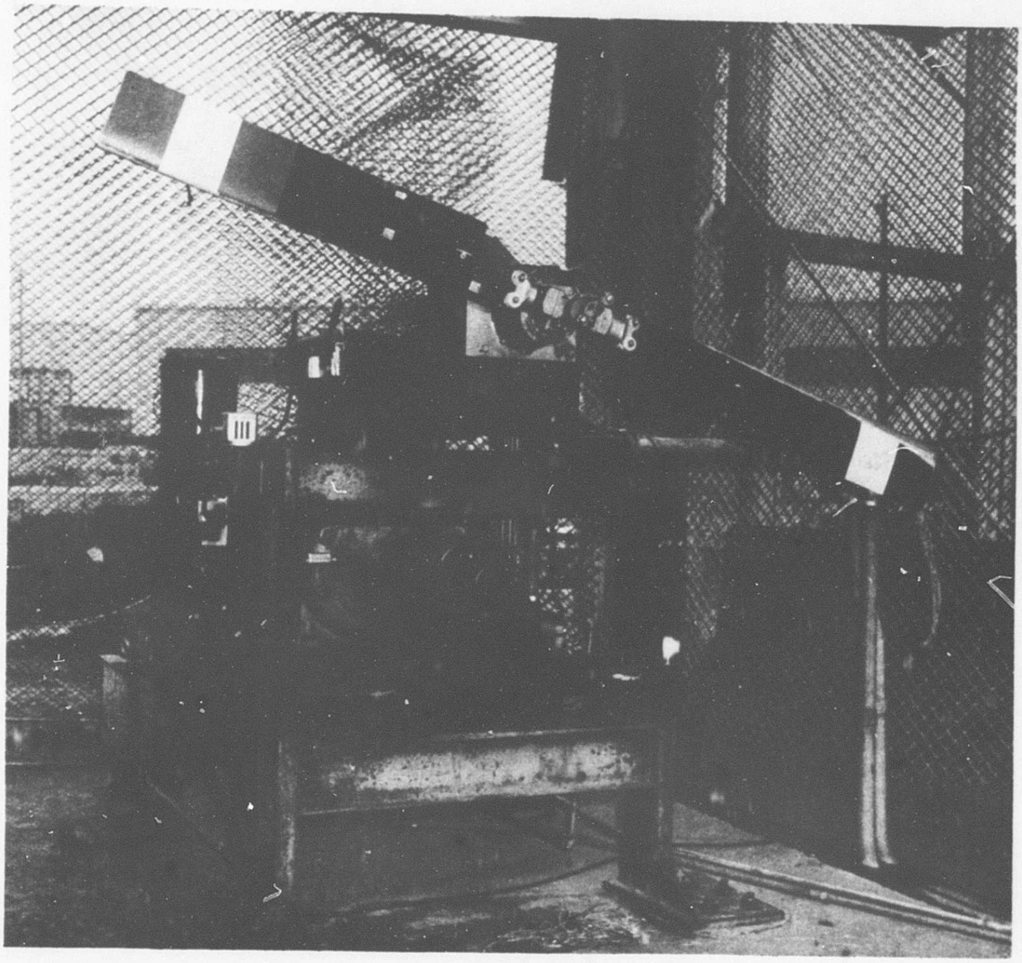
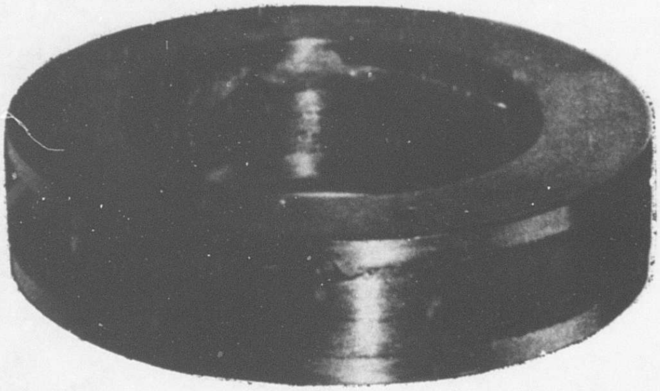
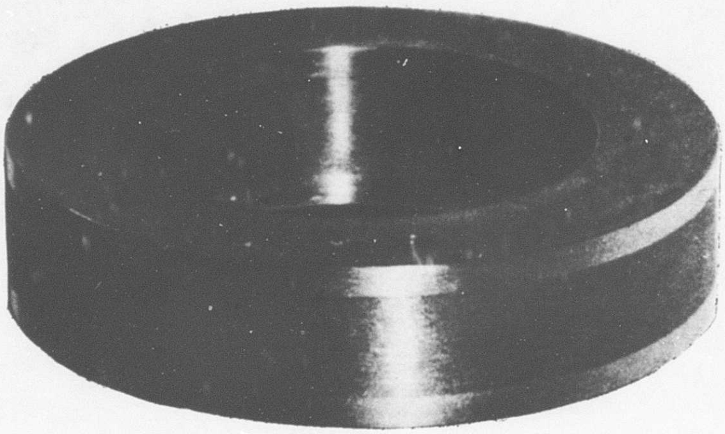


FIGURE 9 . TAIL ROTOR WHIRL STAND.



Outboard Bearing Extruded Elastomer Because of Overload



Inboard Bearing After Same 10-Hour Test

FIGURE 10. BONDED THRUST BEARINGS AFTER WHIRL TEST (CONFIGURATION "BT2").

The principal purpose of the 10-hour whirl tests was to evaluate the thrust bearing configuration; however, two radial bearings (Configuration "BR1") were installed in the flapping hinge during the entire 30 hours of tests. These bearings were in good condition at the completion of tests; however, the flapping amplitudes during whirl test are negligible; therefore, the bearing loads were not representative of flight.

Two preflight whirl tests of .5 hour duration were conducted. The first test was the same tail rotor configuration as the initial 10-hour whirl test but with bearing load equalization and new bearings installed. The second test tail rotor was with Configuration "MR1" radial bearings in the flapping axis and a single Configuration "MT2" thrust bearing in each blade grip. The results of the first test were similar to the earlier tests. The second configuration was found to have a torsional spring rate comparable to the standard UH-1 tail rotor (see Figure 11).

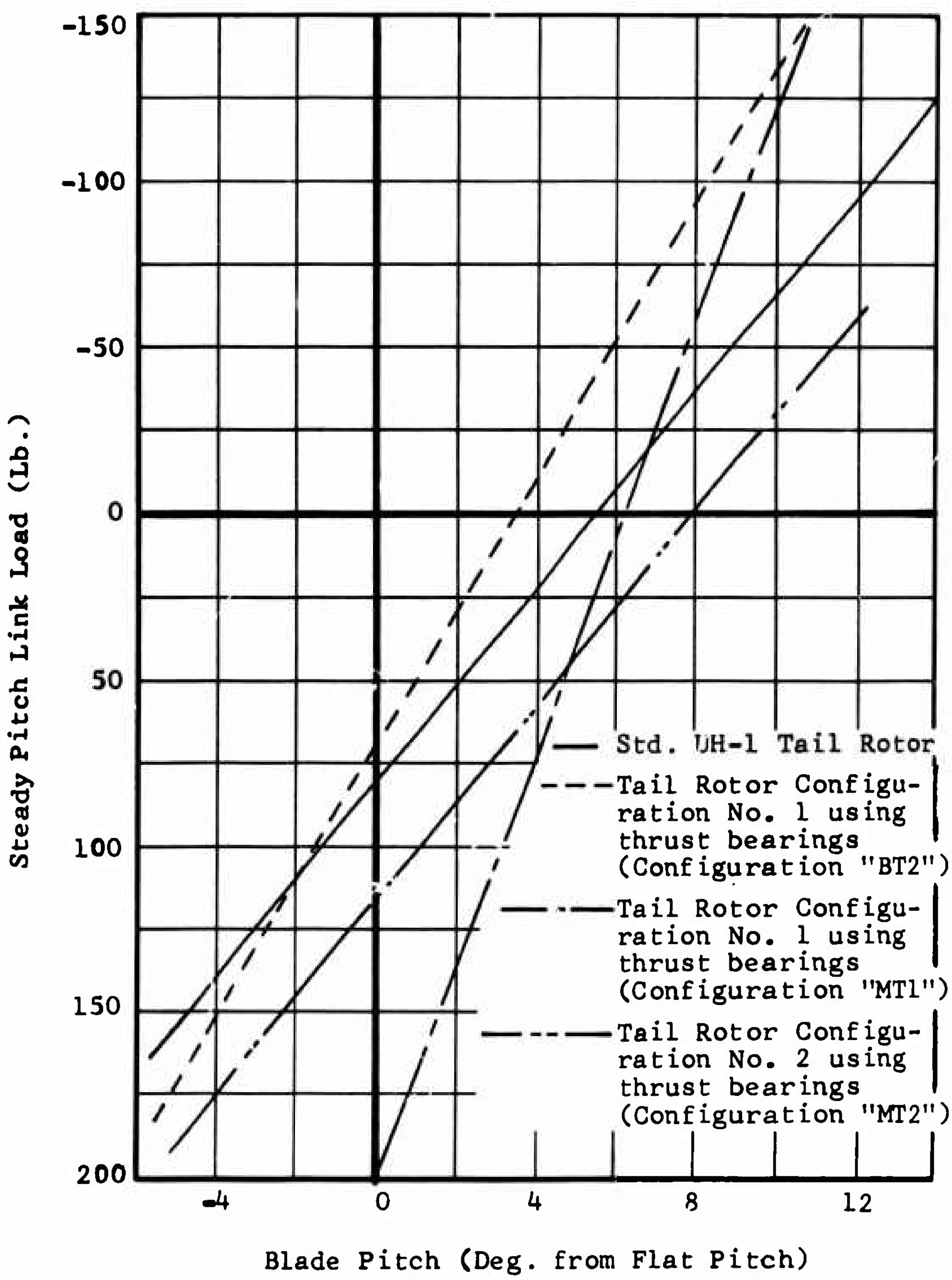


FIGURE 11. WHIRL STAND TAIL ROTOR PITCH LINK LOADS (at 1650 RPM During Pitch Change Cycle).

FLIGHT TEST EQUIPMENT DESCRIPTION

Five tail rotor configurations were flight tested. The two configurations shown in Figure 12 were tested under this program. Three other configurations were tested as a part of Bell Helicopter Company's Independent Research and Development Program.

Standard UH-1B helicopters were used for the flight test. UH-1B helicopter, Serial AF62-2023, was used to test the first configuration, and the remaining tests were conducted using UH-1B, Serial AF-13968. A complete description of the helicopter may be found in Bell Helicopter Company Report 204-947-085, "Detail Specification for UH-1B Utility Helicopter" (Reference 3).

Tail Rotor Configurations

All elastomeric-bearing tail rotor configurations used standard UH-1B tail rotor blades and were 103 inches in diameter. The standard blades have an 0015 NASA airfoil section and a chord of 8.41 inches. The tail rotor was balanced prior to each installation and tracked before each test. The tail rotor assemblies were instrumented as shown in Figure 13, and Figure 14 shows the tail rotor installed on the helicopter.

Tail rotor Configuration 1 used two bonded-type elastomeric thrust bearings in each grip and bonded-type radial bearings in the flapping hinge. Details of this configuration are shown in Figure 12, and an exploded view is shown in Figure 15.

Tail rotor Configuration 2 used a single molded-type elastomeric thrust bearing in each grip and molded-type radial bearings in the flapping hinge. Details of this configuration are shown in Figure 12, and an exploded view is shown in Figure 16.

Tail rotor Configuration 2a is the same as Configuration 2 except that the inboard Teflon journal bearing (572-HES-4-1) has been removed from each grip to provide a lower hub spring rate. Therefore, the blade beam and chord bending loads are transferred to the yoke spindle through the elastomeric thrust bearing. During the bending load transfer, the inboard end of the thrust bearing is free to translate and the bearing face tilts, thus providing the "softest" hub configuration tested during this program.

Tail rotor Configuration 2b is the same as Configuration 2 except that the nut, spacer, and thrust shoulder (572-018-005-5 and 572-HES-24-3 shown in Figure 12) in each grip are replaced by a one-piece part to increase the hub stiffness.

Both Teflon journal bearings were installed in each grip to react the blade bending loads. This is the stiffest hub investigated during this program.

Tail rotor Configuration 2c is the same as Configuration 2b except that the outboard Teflon journal bearing (572-HES-4-1 in Figure 12) is replaced by a rubber bumper (.030-inch clearance is provided between the bumper and the nut). The blade beam and chord bending loads are transferred to the yoke by a face couple on the elastomeric thrust bearing. The calculated spring rate of the thrust bearing face cant (under 16,000-pound compression load) is 1000 inch-pounds per degree. The stiffness of this hub is between that of Configuration 2 and that of Configuration 2a.

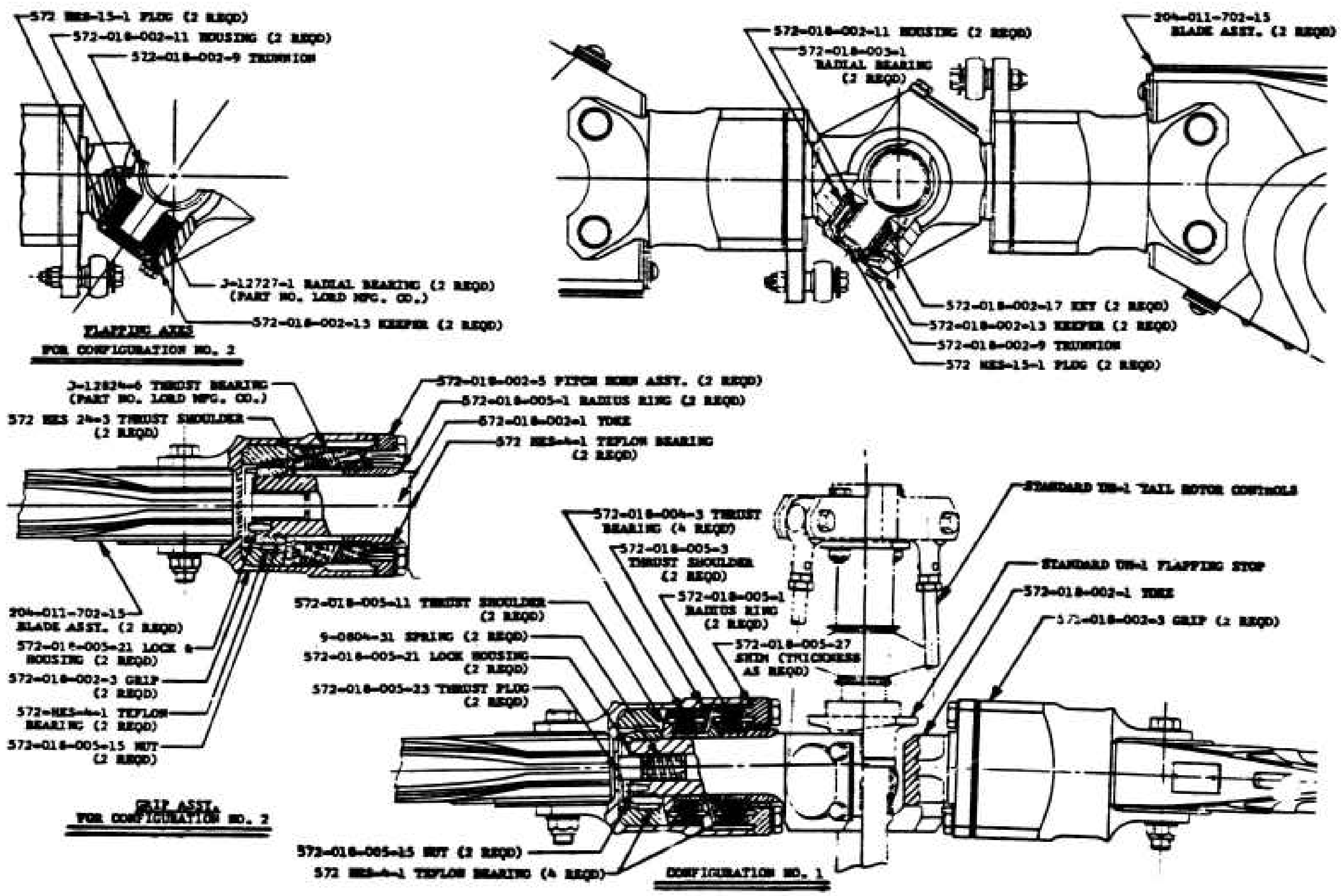


FIGURE 12. ELASTOMERIC BEARING TAIL ROTOR INSTALLATION

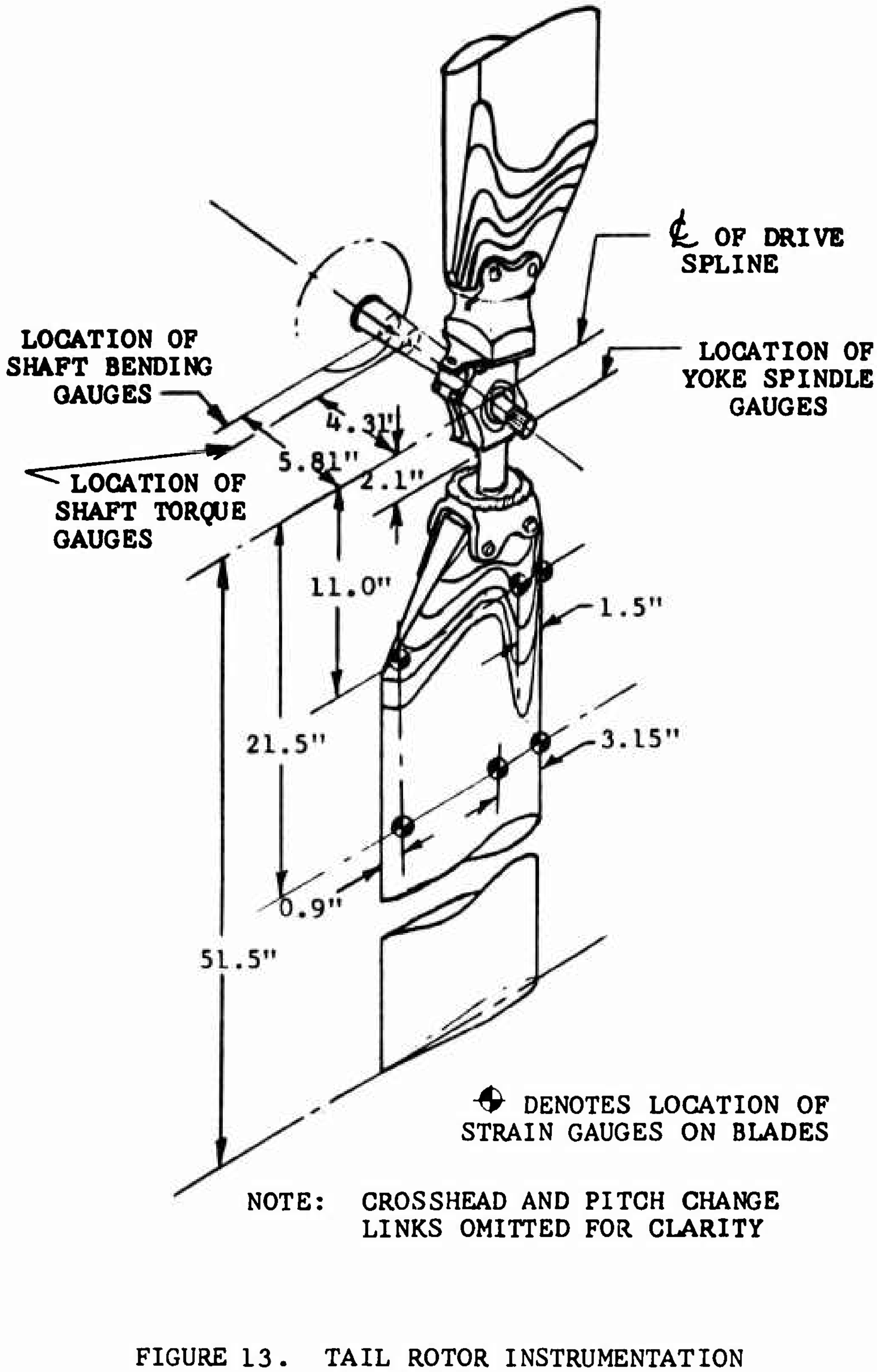


FIGURE 13. TAIL ROTOR INSTRUMENTATION LOCATIONS.

FIGURE 14. ELASTOMERIC BEARING TAIL ROTOR

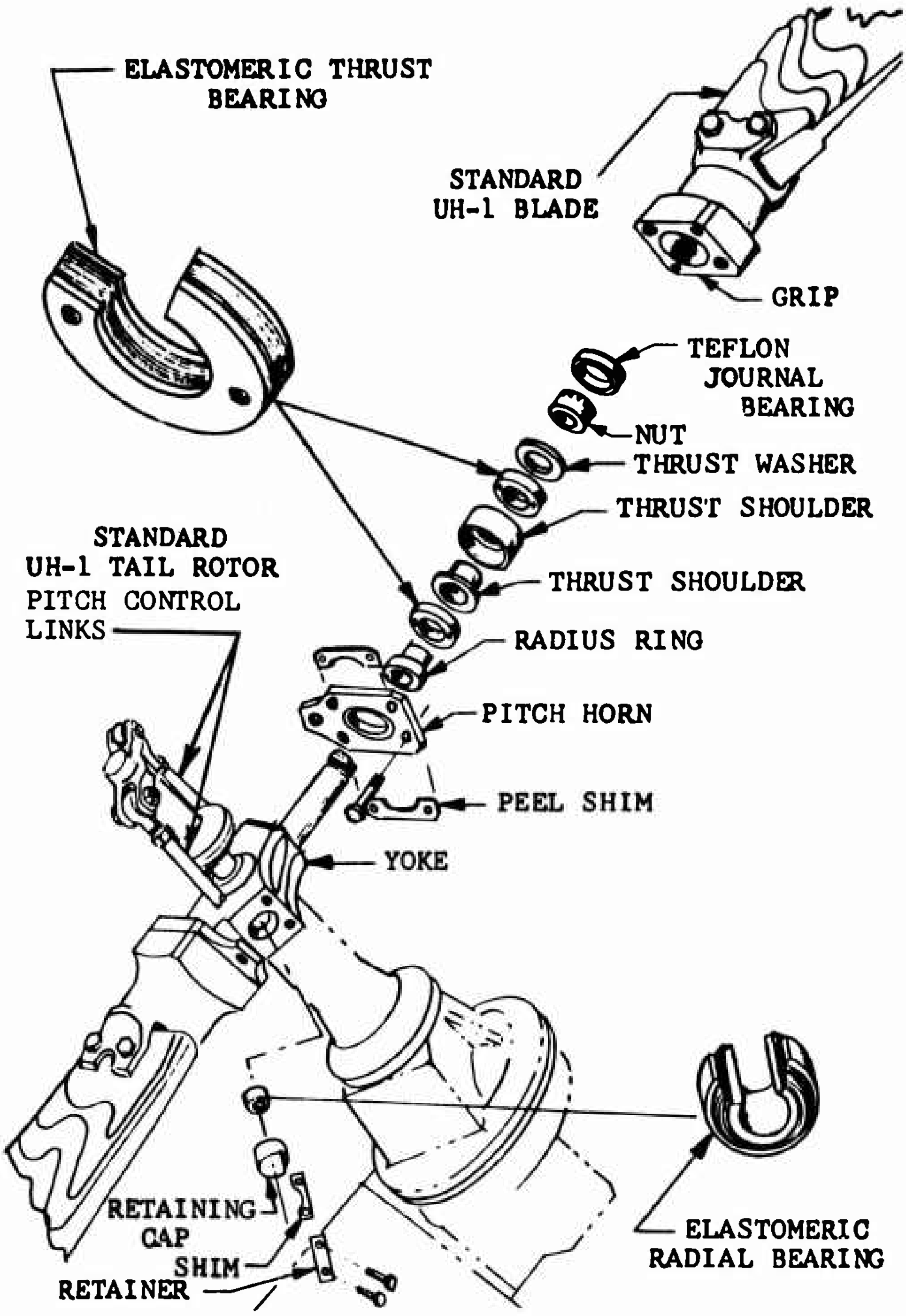


FIGURE 15. ELASTOMERIC BEARING TAIL ROTOR CONFIGURATION 1.

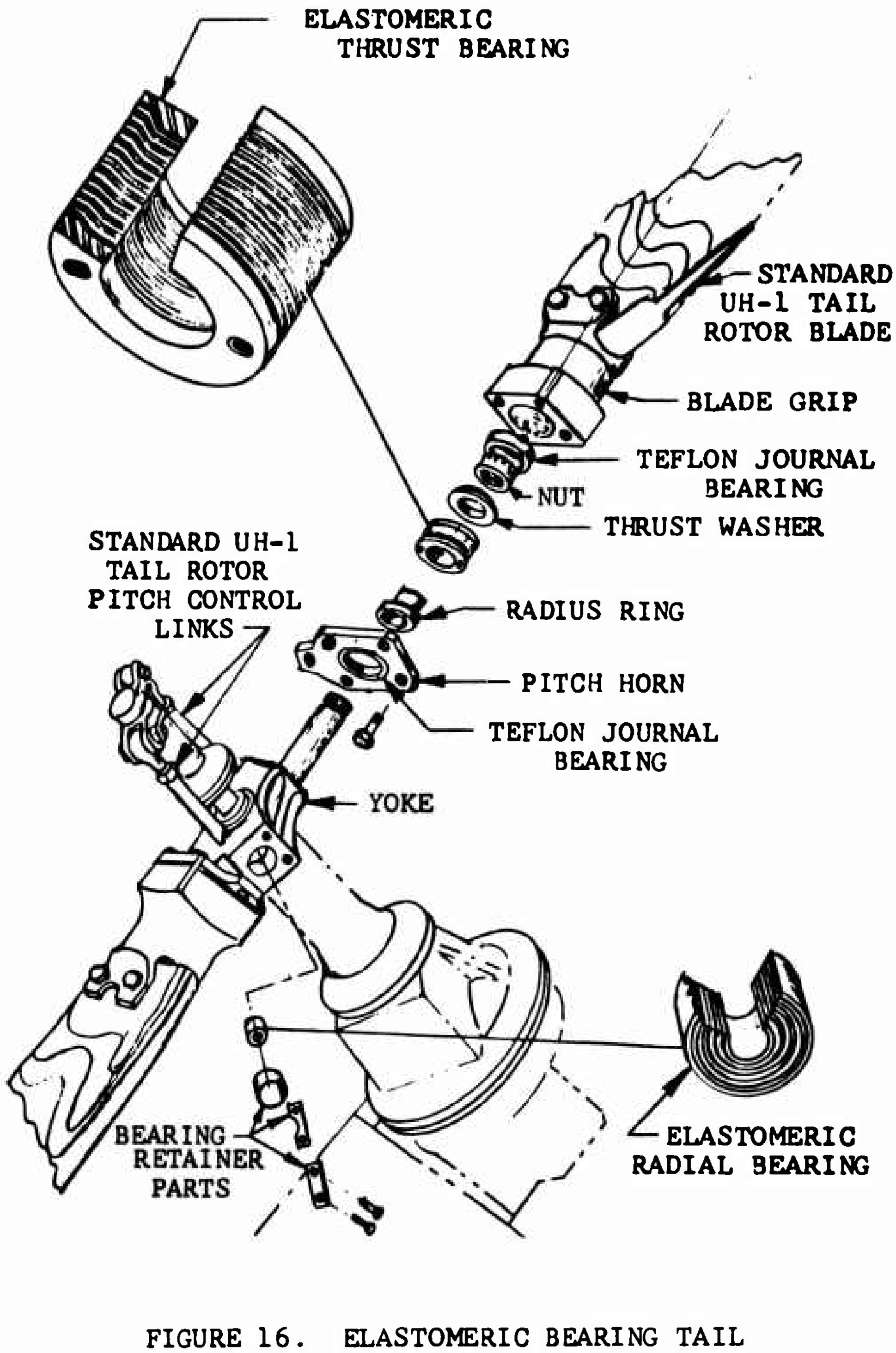


FIGURE 16. ELASTOMERIC BEARING TAIL ROTOR CONFIGURATION 2.

FLIGHT TEST RESULTS

Tabulated flight test results are shown in Appendix I for Configurations 1 and 2. Data are presented graphically for all configurations in Figures 17 through 22 for comparison with each other as well as with standard UH-1 tail rotor data. These plotted data are for tests conducted at 319 main rotor rpm, except for the hover data, which were acquired at 324 main rotor rpm.

Configuration 1, shown in Figure 15, was flight tested 4.3 hours with the helicopter gross weight at 7500 pounds and the center of gravity at neutral (Station 130). Tests were conducted with 1-pound tip-weighted blades installed and with non-tip-weighted blades installed. Both blade configurations were acceptable for boost-on operation; however, the tip-weighted blade configuration was considered to be unacceptable for hydraulic boost-off operation due to high rudder pedal forces. The combined effect of increased centrifugal pitching moment, due to the tip weights, and the torsional spring forces from the two elastomeric thrust bearings per grip caused the steady-control moment which loaded the rudder pedal control system excessively. The loads recorded for this configuration compared favorably with standard UH-1 tail rotor data, except for the higher blade beam oscillatory loads which resulted from the cyclic beam mode's being excited by near resonant forces.

Configuration 1 (non-tip-weighted blades) was also tested during ground run with molded-type elastomeric thrust bearings (Configuration "MT1") installed. The rudder pedal forces required for pitch changes were extremely high during boost-on and boost-off operations,and the test was terminated without acquiring any presentable data.

Configuration 2, shown in Figure 16, was flight tested 2.1 hours using non-tip-weighted blades. During the first part of this test program, the helicopter gross weight was 7500 pounds with a neutral center of gravity (Station 130). The remainder of the flight tests were conducted with the helicopter weight increased to a gross weight of 8600 pounds and the center of gravity moved forward to Station 125.5. Four pilots evaluated this configuration and found the rudder pedal forces to be satisfactory during both normal and boost-off operation.

Configuration 2a was tested 2.6 hours at the same conditions as the first part of Configuration 2 tests. Data from these tests show the tail rotor blade and hub loads to be satisfactory; however, the desired reduction in the oscillatory

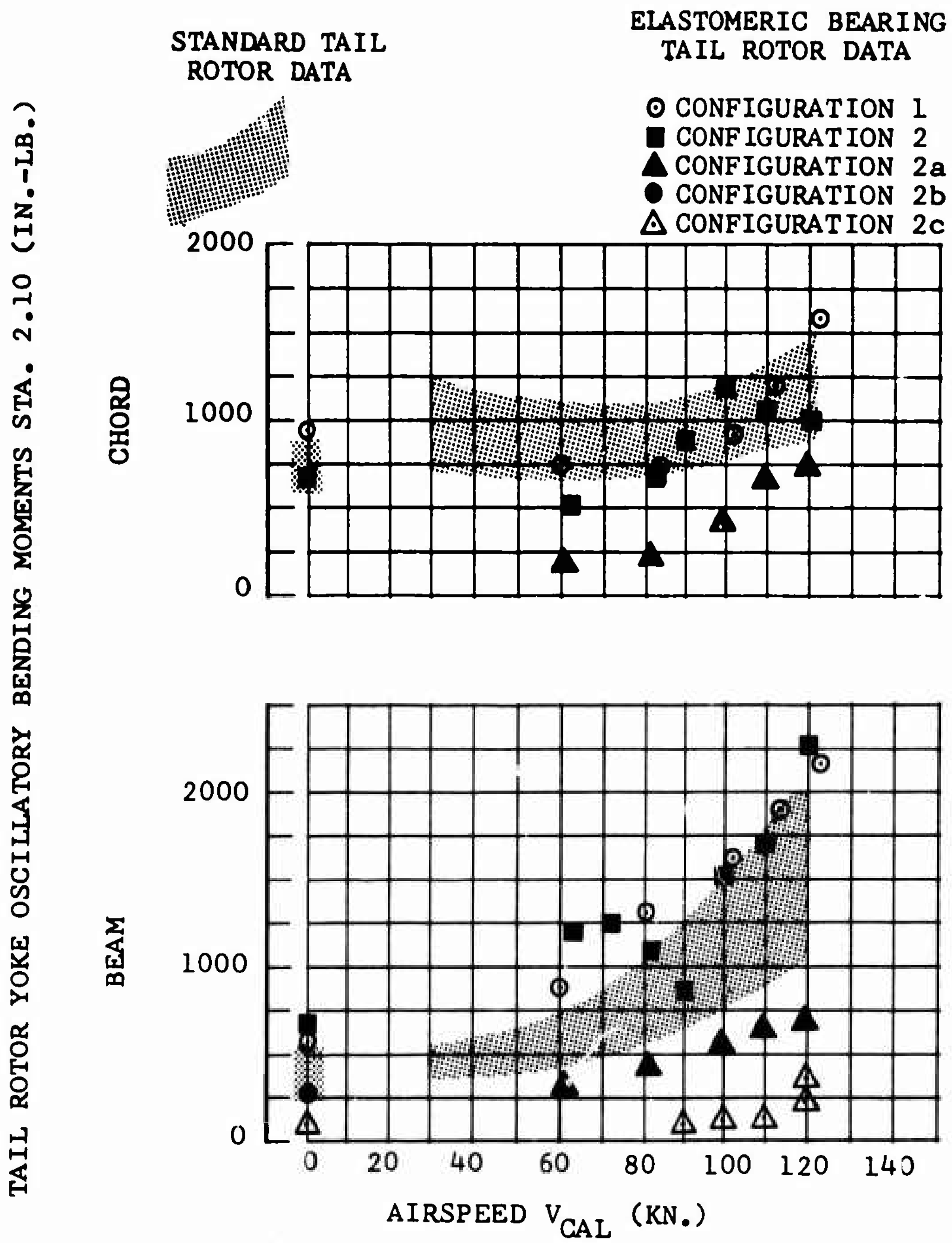
beam bending loads at blade Station 21.5 was not obtained. In addition, tail rotor thrust pulsations were experienced at speeds of 110 and 120 knots. This condition was attributed to hub softness causing an offset hinge, such that an effective negative delta-three due to blade motion with respect to the hub was created. The offset hinge effect, in this configuration, is also believed to have caused the increase in the parallel bending oscillatory loads. Boost-off pedal operating forces were considered to be low by comparison with the standard UH-1 tail rotor.

Configuration 2b was tested .7 hour at the same conditions as Configuration 2a tests. Testing of this configuration was terminated when it was found that apparently all blade loads would be in excess of the data recorded during Configuration 2 tests.

Configuration 2c was tested 1.8 hours at the same conditions as Configuration 2a tests. The yoke oscillatory loads recorded during these tests were very low when compared with standard UH-1 tail rotor data. The blade oscillatory loads recorded were also satisfactory, with the exception of one flight at 120 knots which showed the Station 21.5 blade beam oscillatory loads to be high. This increase in oscillatory load is attributed to high steady blade beam bending loads that bottomed out the bumper on the yoke nut outside diameter, thus changing the load path which moved the tail rotor frequency closer to the helicopter operation resonance. The parallel shaft oscillatory bending load data were higher than the standard UH-1 data for this configuration. However, these loads did not cause a problem, since the tail rotor mast design was determined by other higher load requirements. This is the best configuration tested to date with respect to oscillatory loads as shown in Figures 17 thru 19.

Figure 19 shows the Station 21.5 blade beam oscillatory loads to be slightly higher for all elastomeric bearing tail rotors tested. These high beam oscillatory loads indicate the blade beam natural frequency of the test tail rotor to be near resonance at helicopter operating speeds. However, it is believed that the beam stiffness of the blade can be tailored to eliminate this resonant condition.

Under another program, flight tests were conducted with only the molded radial bearings installed in the flapping hinge of a standard UH-1 tail rotor. Comparison of the results with standard tail rotor results indicated that no measurable changes were made in the tail rotor's dynamic characteristics.



(Chord instrumentation inoperative for Configurations 2b & 2c)

FIGURE 17. TAIL ROTOR YOKE BEAM AND CHORD OSCILLATORY BENDING MOMENTS(STA. 2.10) VERSUS AIRSPEED.

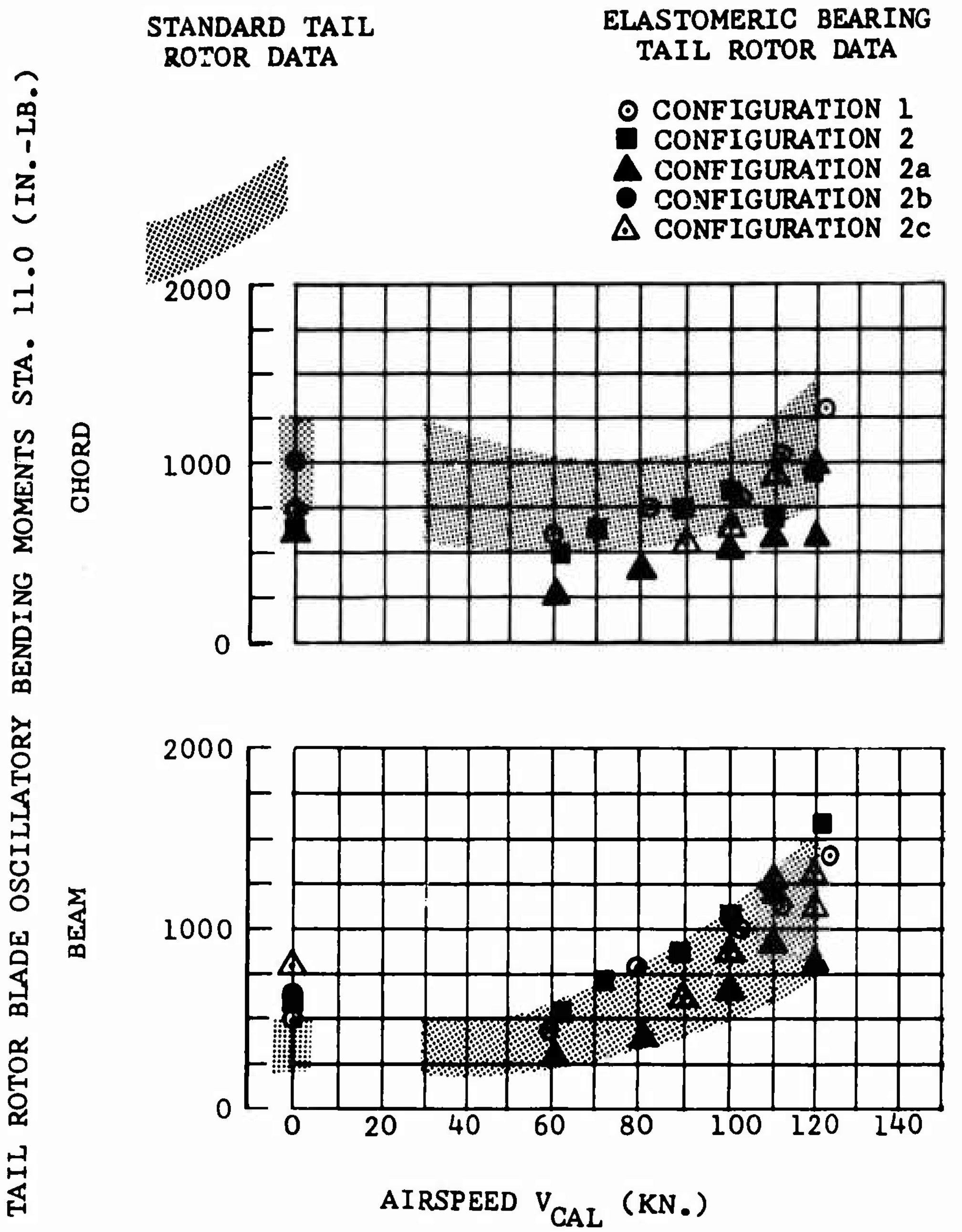


FIGURE 18. TAIL ROTOR BLADE BEAM AND CHORD OSCILLATORY BENDING MOMENTS (STA. 11.0) VERSUS AIRSPEED

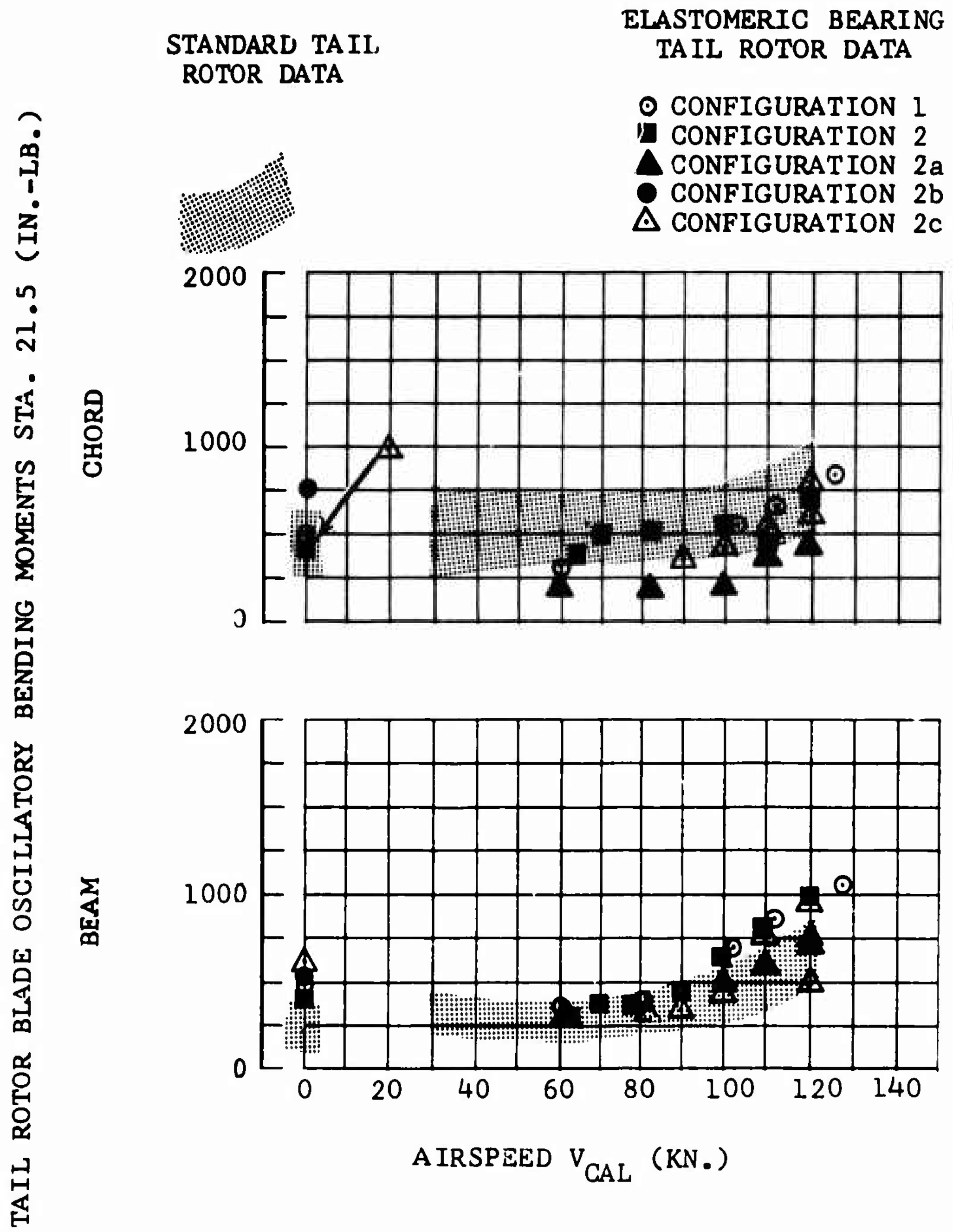


FIGURE 19. TAIL ROTOR BLADE BEAM AND CHORD OSCILLATORY BENDING MOMENTS (STA. 21.5) VERSUS AIRSPEED.

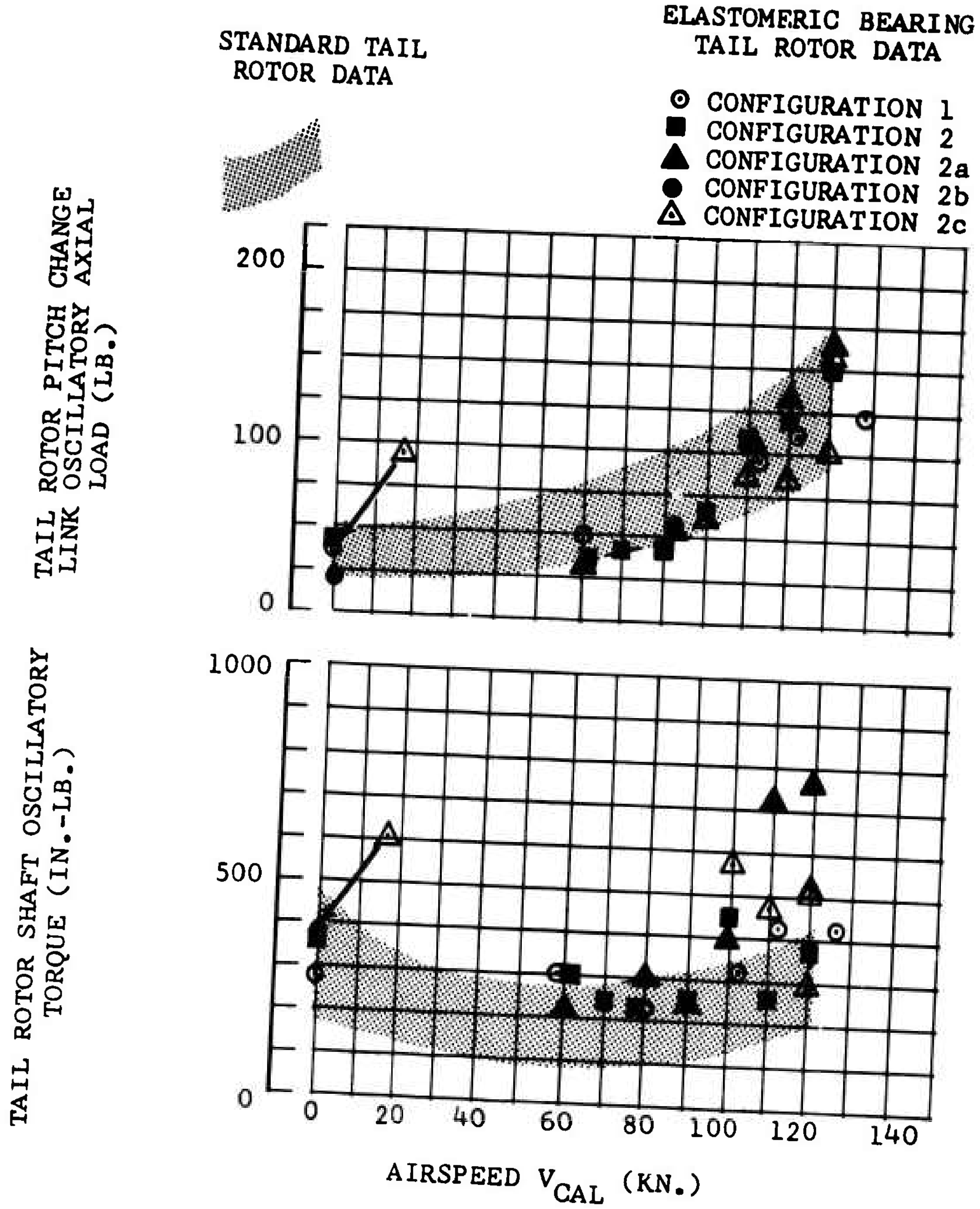


FIGURE 20. TAIL ROTOR SHAFT OSCILLATORY TORQUE AND PITCH CHANGE LINK OSCILLATORY LOAD VERSUS AIRSPEED.

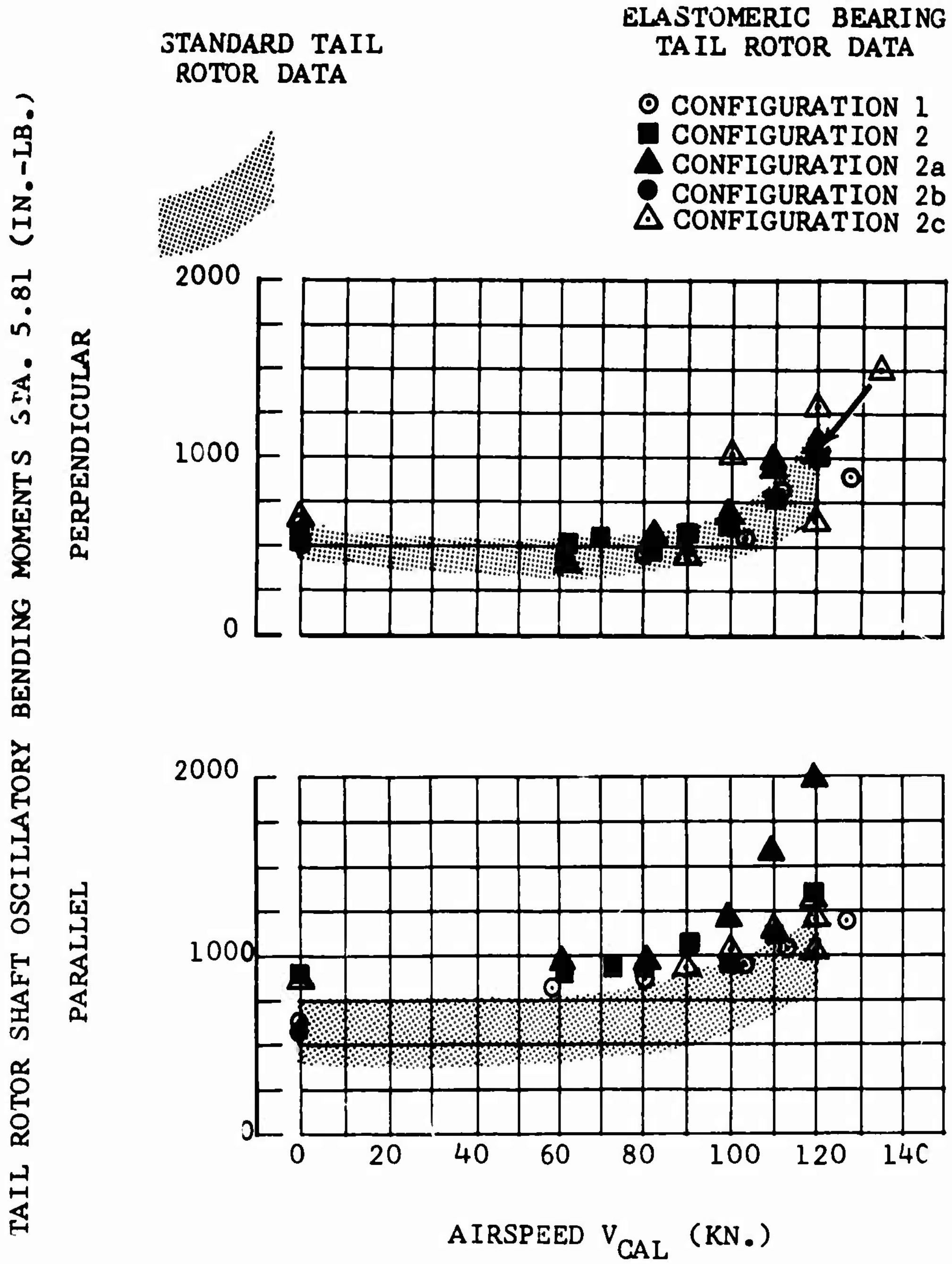


FIGURE 21. TAIL ROTOR SHAFT PARALLEL AND PERPENDICULAR BENDING MOMENTS (STA. 5.81) VERSUS AIRSPEED.

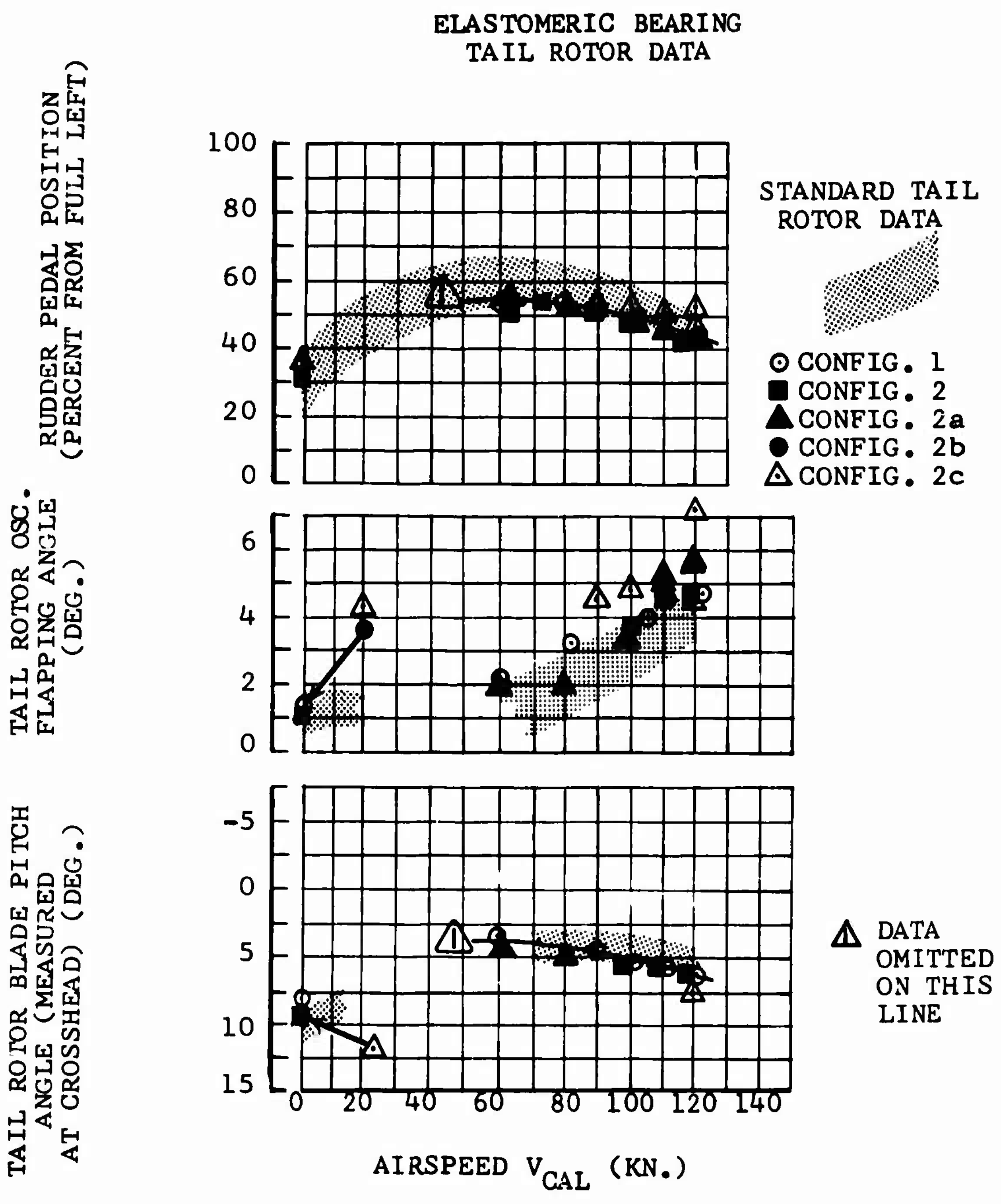


FIGURE 22. TAIL ROTOR BLADE PITCH ANGLE, OSCILLATORY FLAPPING ANGLE AND RUDDER PEDAL POSITION VERSUS AIRSPEED.

TAIL ROTOR DYNAMICS

The dynamics of the elastomeric-bearing tail rotor were evaluated on the basis of frequency calculations, vibration test data, and whirl test data.

SHAKE TEST EVALUATION

Vibration tests were conducted with the Configuration 1 tail rotor to investigate its dynamic characteristics. The tail rotor was installed in a dynamic shake fixture and tested using two Model C-10 MB shakers, as shown in Figure 23.

The test tail rotor assembly was excited over a frequency range of 60 to 3600 cpm using the MB C-10 exciter and an amplifier and control unit. See Table II for pertinent test results.

TABLE II. SHAKE TEST NATURAL FREQUENCIES AND DAMPING FOR MODEL 572 TAIL ROTOR

Mode	Frequency - Cycles Per Minute	Critical Damping - %
1st beam symmetric	942	11.0
2nd beam symmetric	4860	7.3
3rd beam symmetric	14400	3.1
4th beam symmetric	26400	2.0
1st beam asymmetric	3240	Not measured
1st chord symmetric	1620	Not measured

The tail rotor natural frequencies as determined from these tests are compared with the whirl test data in Figures 24 and 25. The natural frequencies as determined by the vibration test data are lower than the frequencies determined by calculations and whirl test results. This is due to the change in load path caused by the blade centrifugal force during the rotating tests.

ROTATIONAL DYNAMIC EVALUATION

The tail rotor frequencies were determined during whirl and ground tests. The rotor natural frequencies are plotted as a function of rotor rpm in Figures 24 and 25. During rpm sweeps (at a constant blade pitch), each mode passes through several excitation sources. These intermediate resonant points are monitored and plotted, and the resulting curves are extrapolated to operating rpm. The rotor natural frequencies (at operating speed) for the various tail rotor configurations are compared in Figure 26. Solid and open block symbols are

used to indicate the location of each mode relative to the excitation sources. The width of the symbols represents the effect of blade pitch change on the natural frequency.

As can be seen in Figure 26, the collective modes are well located with respect to the excitations both in the standard UH-1 rotor and in all elastomeric-bearing rotor configurations. The second collective mode is seen to be strongly affected by bearing stiffness, but in all cases is located between its principal excitation sources two-per-rev and four-per-rev.

The cyclic modes are associated, in most tail rotor designs, with resonant amplification problems. The first mode (solid blocks) is the first symmetric inplane natural frequency; the second mode (open blocks) is the first asymmetric vertical natural frequency of the "S" mode.

The first inplane mode varies considerably with changes in bearing stiffness. It was found that as this mode is "tuned" away from two-per-rev, the oscillatory chord loads are reduced. The ability to locate the inplane mode by altering the bearing geometry is a desirable tool, presently not readily available in the UH-1 tail rotor. The "S" mode did not vary appreciably with changes in elastomeric-bearing hub stiffness. The mode shape is such that maximum bending occurs near midspan, and control of this mode is basically a blade stiffness function. As a result, it was found that the blade three-per-rev oscillatory bending moments remained generally the same regardless of the elastomeric-bearing hub configuration. However, the two-per-rev and four-per-rev reductions achieved by changing the blade to hub stiffness (through the use of elastomeric bearings) did result in an overall midspan oscillatory load reduction.

ANALYTICAL EVALUATION

The dynamic characteristics of the elastomeric-bearing tail rotor were analytically investigated in terms of the individual characteristics of the bonded-type elastomeric bearings, the Teflon journal bearings, and the rotor assembly.

The characteristics of the elastomeric elements will be reflected in the rotor dynamics, but in the subject design the rotor dynamics are also dependent upon the blade root restraint and any damping provided by the Teflon journal bearings. The results of dynamic calculations indicated that the bonded elastomeric bearings would have a negligible effect on the tail rotor natural frequencies. These calculations are given in Appendix II.

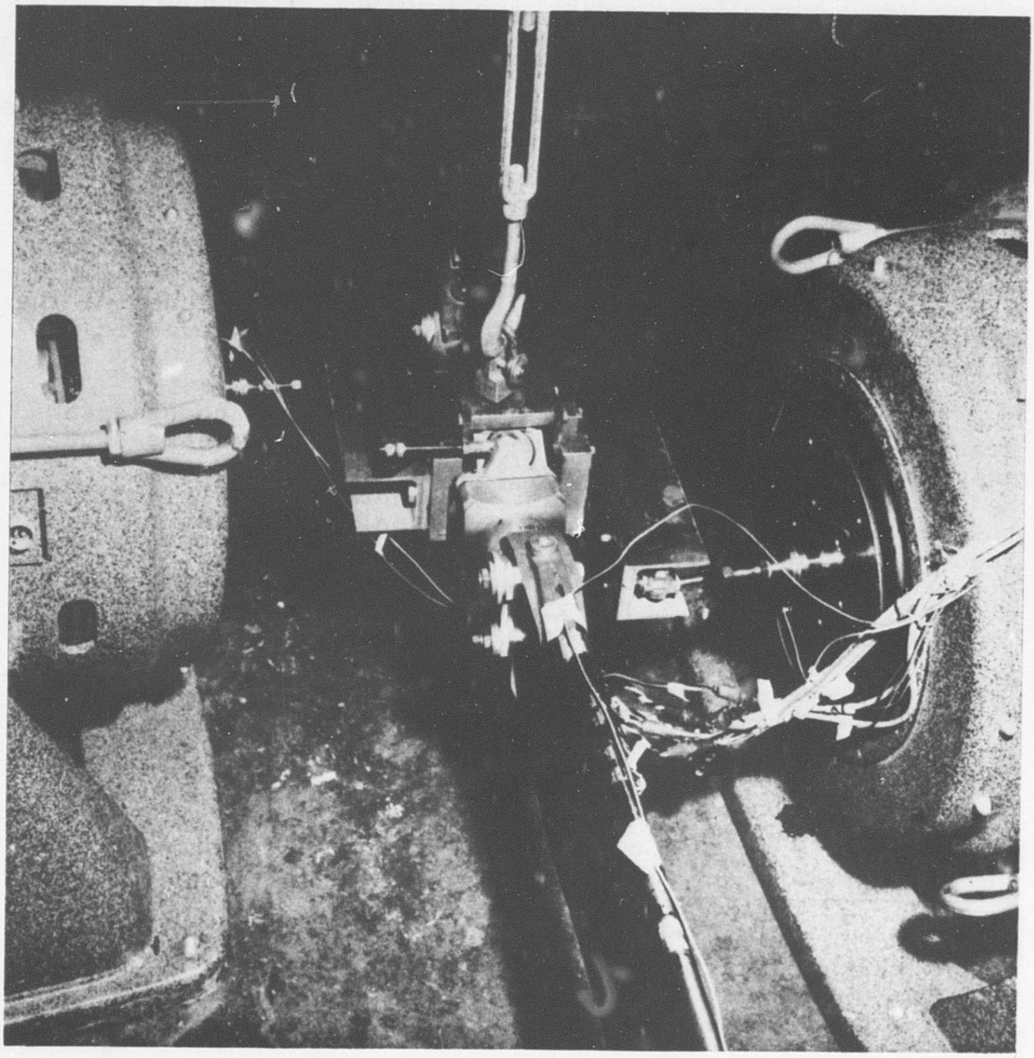


FIGURE 23. VIBRATION SHAKE TEST OF TAIL ROTOR.

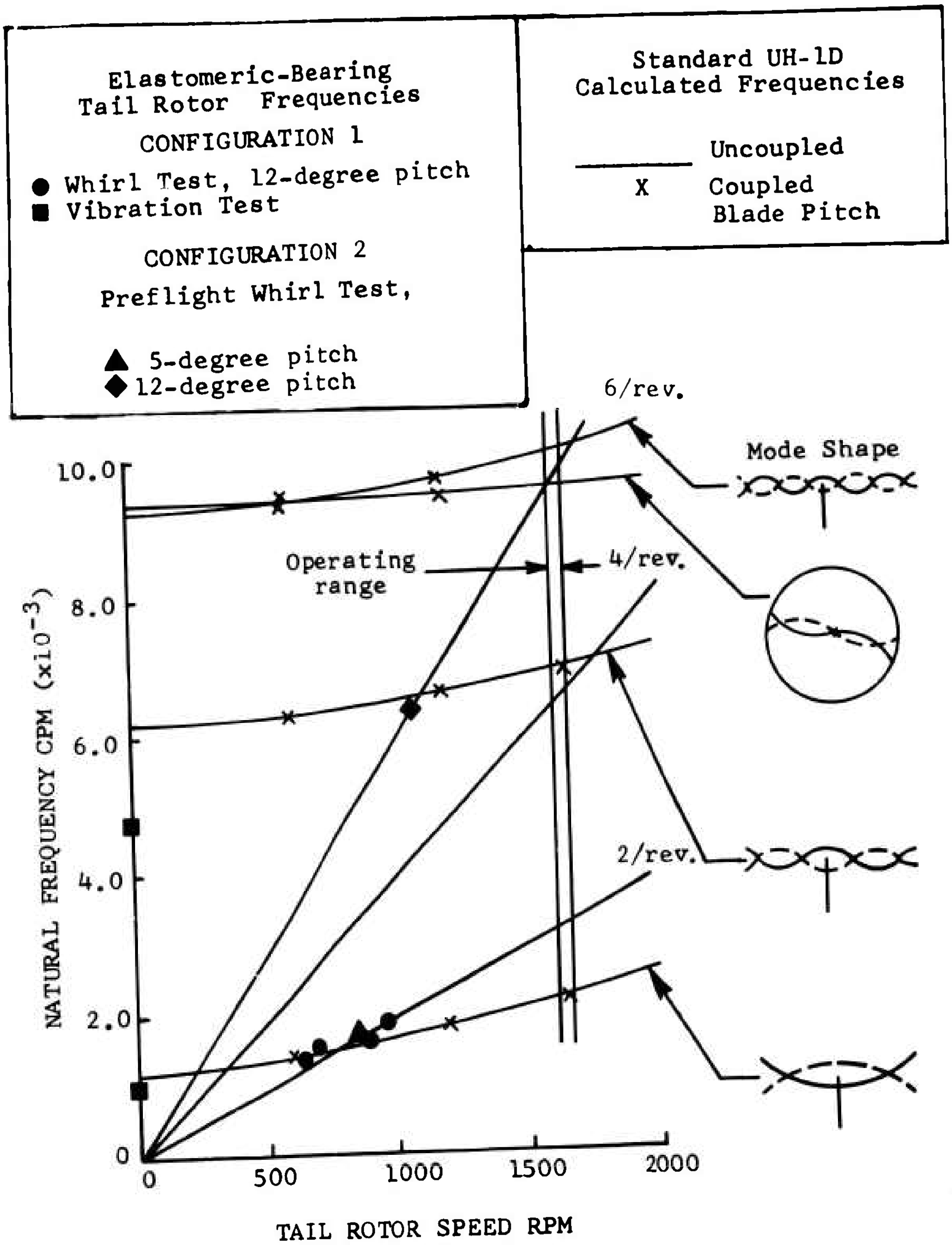


FIGURE 24. NATURAL FREQUENCIES - COLLECTIVE MODES.

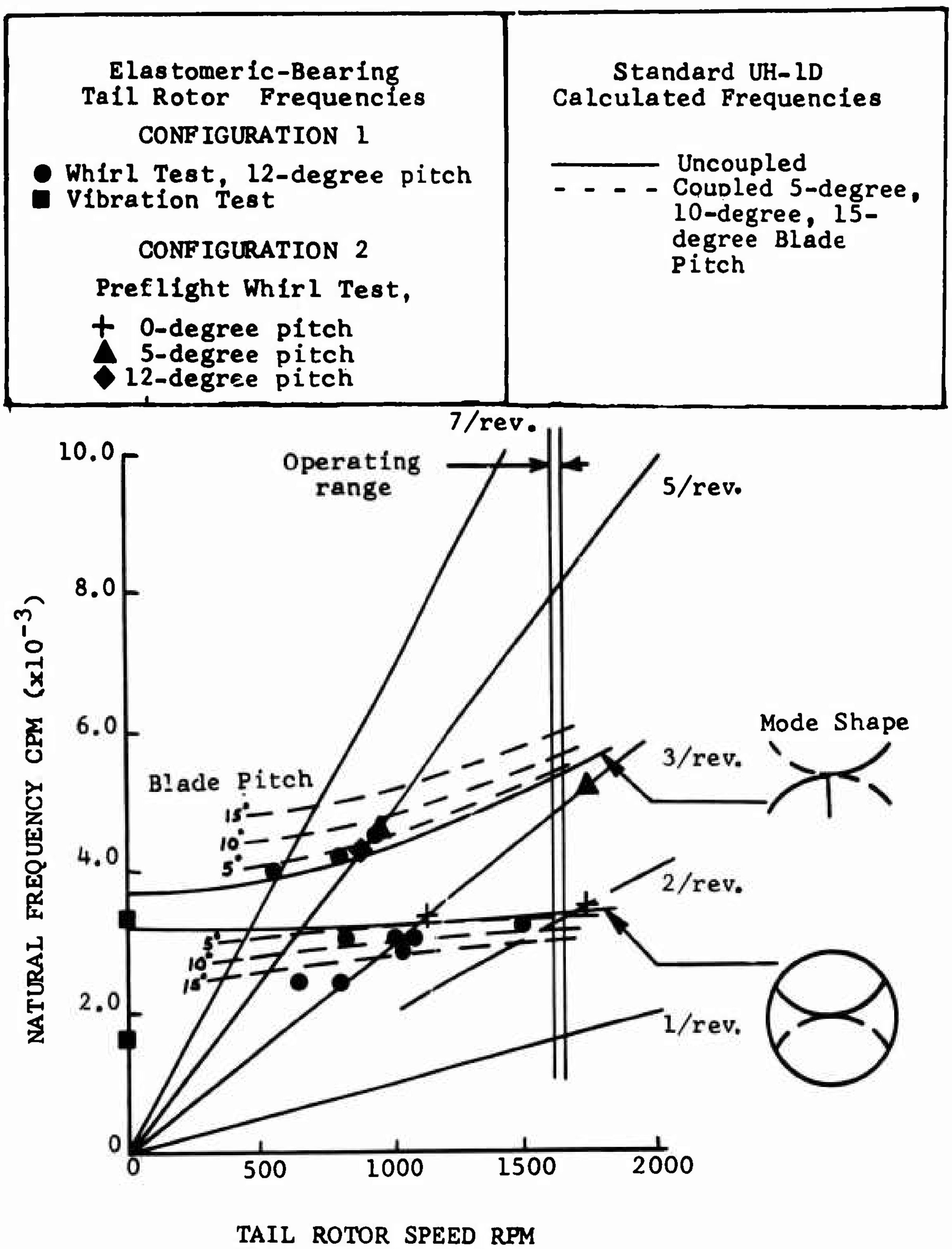


FIGURE 25. NATURAL FREQUENCIES - CYCLIC MODES.

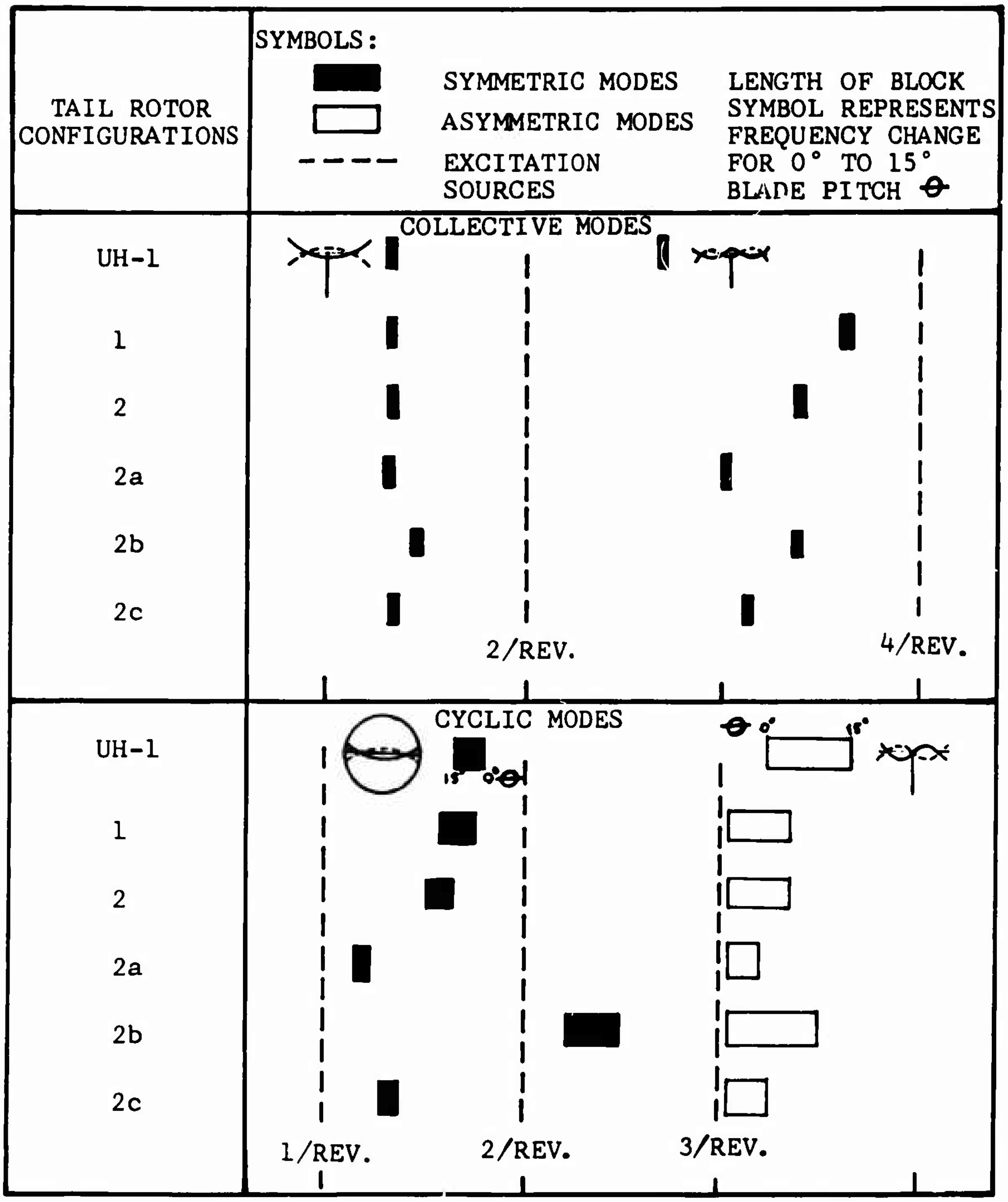


FIGURE 26. COMPARISON OF ROTOR NATURAL FREQUENCIES AT OPERATING SPEED FOR VARIOUS TAIL ROTOR CONFIGURATIONS.

CONCLUSIONS

Bonded Elastomeric Bearings

The bonded elastomeric bearings used during this program were found to be unsuitable for the UH-1 tail rotor application. The radial bearings of this type proved to have inadequate service life, and the torsional spring rate of the thrust bearings was found to be too high for helicopter boost-off operation. Additionally, fabrication difficulties were encountered, which are believed to be inherent in the manufacture of the bonded elastomeric bearings.

Although considerable progress was made during this program in the advancement of the state of the art of this bearing design, major improvements are required before the bonded elastomeric bearings would be suitable for helicopter applications as envisioned at this time. In view of the success with the molded bearings discussed below, it is recommended that no additional work relating to application be undertaken with bonded-type bearings until such time that new processes, elastomers, etc., are developed to overcome the present difficulties.

Molded Elastomeric Bearings

The final configuration of the molded elastomeric thrust and radial bearings used during this program proved to be marginally satisfactory for the UH-1 tail rotor application. Acceptable helicopter boost-off pedal forces and acceptable (but marginal) rotor natural frequencies and structural loadings were shown. It is believed that the operational life of the radial and thrust bearings of this type would be limited by the shelf life of the elastomers ($\sim$ 5 years).

In any rotor pitch change bearing application, the placement of rotor natural frequencies is most important. The elastomeric bearing provides a means by which the designer can alter the load path from the blade to the hub, thus altering the rotor frequency. Although, for any given design, this may require a "tuning" process to achieve correct frequency placement, use of the elastomeric bearing does provide the designer with a useful tool to adjust the rotor frequencies within the same hub geometry. Data provided herein define the effects of such bearings and, consequently, the load path and frequency changes.

It is concluded that further researches should be conducted with the molded elastomeric bearing. This additional work should consist of the design, fabrication, and evaluation of an elastomeric bearing tail rotor using only elastomeric

bearings in the blade grip (no journal bearings) to carry the blade centrifugal force, transfer the blade bending loads to the yoke and provide blade pitch change motions. This blade load path results in an additional tool to effect rotor natural frequency changes. Laboratory and flight test of the new tail rotor configuration will define the ultimate potential of the elastomeric bearing in this rotorcraft application.

REFERENCES

1. Herrick, R. C., Development of the Laminated Elastomeric Bearing Technical Documentary Report, No. ASD-TDR-63-769, Franklin Institute Report F-B1883, Aeronautical Systems Division, Air Force System Command, Wright-Patterson Air Force Base, Ohio, August 1963.

2. Proposal for Flight Test Evaluation of Laminated Elastomeric Bearings in the UH-1 Helicopter Tail Rotor, BHC Report 299-099-277, Bell Helicopter Company, Fort Worth, Texas, March 1965.

3. Hatton, R. R., Detail Specification for UH-1B Utility Helicopter, BHC Report 204-947-085, Bell Helicopter Company, Fort Worth, Texas, May 1961.

APPENDIX I
TABULATED FLIGHT TEST RESULTS

TABLE III. INSTRUMENTATION

The following channels of tail rotor instrumentation were recorded on an oscillograph during the ground run and flight test program. Figure 13 presents the locations of the instrumentation strain gauge bridges on the tail rotor shaft, yoke, and blade. For reference, the instrumented blade is identified as the "red" blade.

CHANNEL	SIGN CONVENTION FOR POSITIVE VALUES	UNITS
Yoke Spindle Sta. 2.1, Chord Bending	Blade leading edge in tension	in./lb.
Yoke Spindle Sta. 2.1, Beam Bending	Blade bends toward tail boom	in./lb.
Blade Sta. 11.0, Chord Bending	Blade leading edge in tension	in./lb.
Blade Sta. 11.0, Beam Bending	Blade bends toward tail boom	in./lb.
Blade Sta. 21.5, Chord Bending	Blade leading edge in tension	in./lb.
Blade Sta. 21.5, Beam Bending	Blade bends toward tail boom	in./lb.
Pitch Link (Red) Axial Load	Tension	lb.
Shaft Torque	Blade leading edge in tension	in./lb.
Shaft Perpendicular Bending	Side of shaft toward leading edge of instrumented blade in tension	in./lb.
Shaft Parallel Bending	Shaft bends toward instrumented blade	in./lb.
Flapping Position	Instrumented blade toward tail boom	deg.

TABLE III. - Continued

CHANNEL	SIGN CONVENTION FOR POSITIVE VALUES	UNITS
Rudder Pedal Position	Right rudder	Pct. from full left rudder
Blade Angle*	Leading edge toward tail boom	deg.

*NOTE: The blade angle was measured at the pitch change crosshead and will not monitor the effect of δ_3 (pitch change with flapping).

TABLE IV. GROUND RUN AND FLIGHT LOG

Date 1966	G.R. or Flt. No.	Table No.	T/Rotor Configuration	Test Conducted
2 June	218B	V	No. 1 (1-lb. tip weight ea. blade)	Climb, Level Flt.76-124 Kn., Turns, Auto-rotate
9 June	G.R.73	VI	No. 1 (non-tip weighted blades)	324 M/R RPM
10 June	222A	VII	No. 1 (non-tip-weighted blades)	Hover, Turns, Accel., Decel., Level Flt.
10 June	222C	VIII	No. 1 (non-tip-weighted blades)	Climb, Level Flt.60-124 Kn., Turns, Auto-rotate
18 November	G.R. 32A	IX	No. 2 (std. blades)	Engine Idle to 324 M/R RPM
22 November	172A	X	No. 2 (std. blades)	Level Flt. 47-121 Kn., Turns, Hover, Hover Turns, Side-ward Flt.
28 November	173A	XI	No. 2 (std. blades)	Hover, Side-ward Flt., Accel.,Decel., Level Flt. 100-120 Kn., @ a High G.W. of 8606 Lb. C. of G. of 125.5

TABLE V. TAIL ROTOR LOADS AND DISPLACEMENT DATA

Model UH-1B 543 Flt. 218-B Date 2 June 1966
Ship AF62-2023 Tail Rotor Configuration Number 1

CTR NO.	TEST CONDITION (Units Explained in Fig. 13 & Table III.)	MAIN ROTOR RPM	V_{CAL} (KN.)	T/R YOKE CHORD AT 2.1	
				MEAN	OSC.
795	Full Power Climb	319	76.5	-621	1209
796	Level Flight	319	60.0	-768	735
797	Level Flight	319	93.0	-850	719
798	Level Flight	319	103.0	-784	948
799	Level Flight	319	112.5	-719	1275
800	Level Flight	319	124.5	-899	1651
801	Left Turn	319	103.0	-539	1062
802	Right Turn	319	103.0	-899	997
803	Autorotation	319	73.0	-1046	752

CTR NO.	T/R YOKE BEAM AT 2.1		T/R BLADE CHORD AT 11.0		T/R BLADE BEAM AT 11.0	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
795	-1078	1491	332	1093	-169	847
796	-3947	1009	142	618	-1282	459
797	-2639	1491	237	618	-435	822
798	-2478	1560	190	855	-387	822
799	-1652	1973	523	1093	-411	1040
800	-1675	1904	95	1331	-121	1427
801	-2501	1583	475	951	-290	919
802	-2134	1858	-142	808	-943	847
803	-5989	803	-618	618	-2226	435

TABLE V. - Continued

CTR NO.	T/R BLADE CHORD AT 21.5		T/R BLADE BEAM AT 21.5		T/R PITCH LINK RED	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
795	449	748	-248	626	27	113
796	119	419	-606	288	55	35
797	269	449	-358	517	75	75
798	239	599	-338	517	88	88
799	389	748	-427	865	83	123
800	479	838	-268	985	98	123
801	479	658	-388	646	45	106
802	149	569	-407	626	118	118
802	0	419	-945	208	113	37

CTR NO.	T/R SHAFT TORQUE		T/R SHAFT PERP. BEND		T/R SHAFT PARA. BEND	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
795	1729	600	-264	1027	179	1257
796	864	229	117	469	149	808
797	864	335	293	528	209	928
798	900	405	58	704	269	868
799	1023	423	146	792	89	988
800	1076	653	146	792	179	1138
801	1359	582	-29	616	389	1048
802	935	511	234	587	119	1198
802	388	494	0	528	299	718

CTR NO.	T/R HUB FLAPPING		PEDAL POSITION		T/R BLADE ANGLE	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
795	-	3.6	47	-	6.4	-
796	-	1.3	62	-	3.0	-
797	-	3.3	58	-	4.6	-
798	-	4.0	57	-	4.8	-
799	-	4.2	54	-	5.0	-
800	-	4.7	54	-	5.4	-
801	-	4.2	50	-	6.0	-
802	-	3.8	64	-	3.0	-
803	-	0.8	74	-	0.4	-

TABLE VI. TAIL ROTOR LOADS AND DISPLACEMENT DATA

Model UH-1B 543 — Flt. G.R. 73 — Date 9 June 1966
Ship AF62-2023 — Tail Rotor Configuration Number 1

CTR NO.	TEST CONDITION (Units Explained in Fig. 13 & Table III.)	MAIN ROTOR RPM	V_{CAL} (KN.)	T/R YOKE CHORD AT 2.1 MEAN	T/R YOKE CHORD AT 2.1 OSC.
867	Stabilized RPM	324	0.0	-388	1095
868	Boost-Off Slow Pedal Kick	324	0.0	522	1534
869	Shutdown		0.0	-472	1011

CTR NO.	T/R YOKE BEAM AT 2.1 MEAN	T/R YOKE BEAM AT 2.1 OSC.	T/R BLADE CHORD AT 11.0 MEAN	T/R BLADE CHORD AT 11.0 OSC.	T/R BLADE BEAM AT 11.0 MEAN	T/R BLADE BEAM AT 11.0 OSC.
867	-3869	973	1612	830	-789	789
868	-308	1020	2101	1319	-929	929
869	-1068	2160	1563	781	-92	1301

CTR NO.	T/R BLADE CHORD AT 21.5 MEAN	T/R BLADE CHORD AT 21.5 OSC.	T/R BLADE BEAM AT 21.5 MEAN	T/R BLADE BEAM AT 21.5 OSC.	T/R PITCH LINK RED MEAN	T/R PITCH LINK RED OSC.
867	-28	537	-260	573	-38	33
868	367	764	-417	771	-91	56
869	84	481	-125	917	7	28

CTR NO.	T/R SHAFT TORQUE MEAN	T/R SHAFT TORQUE OSC.	T/R SHAFT PERP. BEND MEAN	T/R SHAFT PERP. BEND OSC.	T/R SHAFT PARA. BEND MEAN	T/R SHAFT PARA. BEND OSC.
867	1202	226	-87	726	88	968
868	3015	888	-38	871	381	1085
869	836	906	87	551	117	762

TABLE VI. - Continued

CTR NO.	T/R HUB FLAPPING		PEDAL POSITION		T/R BLADE ANGLE	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
867	-	1.7	45	-	5.9	-
868	-	2.2	24	-	10.1	-
869	-	3.9	52	-	4.5	-

TABLE VII. TAIL ROTOR LOADS AND DISPLACEMENT DATA

Model UH-1B 543 Ship AF62-2023 | Flt. 222-A | Date 10 June 1966 | Tail Rotor Configuration Number 1

CTR NO.	TEST CONDITION (Units Explained in Fig. 13 & Table III.)	MAIN ROTOR RPM	V_{CAL} (KN.)	T/R YOKE CHORD AT 2.1	
				MEAN	OSC.
875	Hover I.G.E.	324	0.0	80	950
876	Hover I.G.E.	314	0.0	112	950
877	Hover I.G.E.	304	0.0	128	901
878	Hover to Left Turn	324	0.0	-144	1497
879	Hover to Right Turn	324	0.0	48	2786
880	Accel. 0 to 37	324	0.0	-322	1513
881	Decel. 37 to 0	324	37.0	-579	1739
882	Level Flight	324	45.6	-676	1159
883	Level Flight	324	53.5	-708	966
884	Level Flight	324	60.0	-692	853
885	Hover Boost-Off	324	0.0	225	1288

CTR NO.	T/R YOKE BEAM AT 2.1		T/R BLADE CHORD AT 11.0		T/R BLADE BEAM AT 11.0	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
875	-1560	642	1355	774	0.0	521
876	-1216	803	1500	629	130	477
877	-826	878	1307	726	0.0	521
878	-688	1147	1258	1258	673	716
879	-2593	1354	1355	2323	-738	1389
880	-1813	1262	1113	1210	130	912
881	-2868	1032	919	1403	-521	912
882	-3235	895	1016	919	-651	651
883	-3580	872	919	726	-738	477
884	-3442	1147	1113	726	-629	586
885	-1239	963	1549	1065	65	629

TABLE VII. - Continued

CTR NO.	T/R BLADE CHORD AT 21.5		T/R BLADE BEAM AT 21.5		T/R PITCH LINK RED	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
875	-111	500	71	479	-58	37
876	0.0	500	61	449	-65	40
877	-27	472	81	428	-68	37
878	83	806	275	561	-88	113
879	-194	1528	-234	1174	25	116
880	-277	778	-20	918	0.0	65
881	-416	917	-275	724	17	42
882	-305	583	-398	479	32	32
883	-389	444	-408	408	32	27
884	-250	416	-428	469	52	42
885	-55	722	-10	684	-73	47

CTR NO.	T/R SHAFT TORQUE		T/R SHAFT PERP. BEND		T/R SHAFT PARA. BEND	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
875	2012	277	0.0	625	384	622
876	2029	364	-28	654	563	622
877	2255	624	28	711	326	741
878	3070	468	227	1194	622	1215
879	1492	416	199	1223	563	741
880	1682	260	142	881	503	681
881	1179	312	113	796	326	800
882	1093	260	113	625	355	829
883	1023	225	56	569	414	711
884	1023	294	56	625	444	800
885	2914	312	-28	825	296	1067

TABLE VII. - Continued

CTR NO.	T/R HUB FLAPPING		PEDAL POSITION		T/R BLADE ANGLE	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
875	-	1.2	33	-	7.6	-
876	-	1.3	31	-	8.2	-
877	-	1.5	29	-	8.2	-
878	-	3.1	21	-	9.8	-
879	-	5.8	44	-	6.0	-
880	-	2.2	40	-	6.4	-
881	-	1.8	52	-	4.6	-
882	-	0.7	56	-	3.8	-
883	-	0.7	57	-	3.4	-
884	-	1.1	57	-	3.4	-
885	-	1.6	16	-	10.2	-

TABLE VIII. TAIL ROTOR LOADS AND DISPLACEMENT DATA

Model UH-1B 543 | Flt. 222-C | Date 10 June 1966
Ship AF62-2023 | Tail Rotor Configuration Number 1

CTR NO.	TEST CONDITION (Units Explained in Fig. 13 & Table III.)	MAIN ROTOR RPM	V_{CAL} (KN.)	T/R YOKE CHORD AT 2.1	
				MEAN	OSC.
902	Full Power Climb	319	67.5	-305	1079
903	Level Flight	319	60.0	-611	740
904	Level Flight	319	82.0	-499	724
905	Level Flight	319	103.0	-450	934
906	Level Flight	319	112.5	-499	1175
907	Level Flight	319	124.5	-547	1610
908	Left Turn	319	103.0	-161	1223
909	Right Turn	319	103.0	-402	1143
910	Autorotation	319	73.0	-1046	789
911	Auto Left Turn	319	73.0	-853	1014
912	Auto Right Turn	319	73.0	-1062	1481
913	Sideward Flight to Left	319	30.0	-1304	1175
914	Sideward Flight to Right	319	30.0	1127	1320
915	Hover I.G.E. to Left Turn	319	0.0	1062	1674
916	Hover I.G.E. to Right Turn	319	0.0	-450	1997

CTR NO.	T/R YOKE BEAM AT 2.1		T/R BLADE CHORD AT 11.0		T/R BLADE BEAM AT 11.0	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
902	-1101	1285	2312	963	716	977
903	-3213	918	2168	626	-195	456
904	-2685	1308	2168	722	86	781
905	-1904	1583	2168	819	412	977
906	-1468	1881	3120	1060	716	1150
907	-1124	2134	2071	1300	694	1389
908	-1445	1721	2312	1060	412	1064
909	-1767	2088	2216	963	217	998
910	-5393	986	1445	674	-1150	499
911	-4911	1055	1686	915	-1194	673
912	-6012	1055	1638	1156	-1585	803
913	-4153	1354	1445	963	-694	912
914	68	895	3276	1060	1476	477
915	-940	1078	2987	1445	1346	738
916	-1767	1354	2505	1541	781	912

TABLE VIII. - Continued

CTR NO.	T/R BLADE CHORD AT 21.5		T/R BLADE BEAM AT 21.5		T/R PITCH LINK RED	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
902	390	614	408	571	-7	73
903	362	307	30	357	42	48
904	223	390	142	408	45	55
905	195	530	142	673	27	93
906	335	670	387	837	30	106
907	279	837	367	1041	12	118
908	474	698	285	796	-22	108
909	195	642	275	684	53	98
910	-307	418	-326	285	106	20
911	-167	558	-204	551	63	32
912	-55	725	-469	653	111	40
913	55	614	-255	663	227	55
914	1005	725	724	561	-197	70
915	837	1005	612	551	-106	126
916	558	1005	428	816	37	128

CTR NO.	T/R SHAFT TORQUE		T/R SHAFT PERP. BEND		T/R SHAFT PARA. BEND	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
902	1226	570	142	711	237	948
903	708	293	-56	455	237	829
904	742	259	-113	455	207	859
905	881	328	-85	540	148	978
906	1053	431	28	768	237	1007
907	1071	414	-256	938	59	1244
908	1174	483	85	711	326	1274
909	811	431	-227	682	385	1333
910	691	345	-28	483	326	800
911	794	552	-28	597	118	1067
912	725	552	-28	768	237	770
913	-345	310	-170	455	118	711
914	3886	673	28	597	266	1096
915	3455	587	142	1109	266	1155
916	3645	673	28	1223	207	859

TABLE VIII. - Continued

CTR NO.	T/R HUB FLAPPING		PEDAL POSITION		T/R BLADE ANGLE	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
902	-	2.5	38	-	6.2	-
903	-	1.1	54	-	2.8	-
904	-	2.3	54	-	3.0	-
905	-	3.2	49	-	4.2	-
906	-	4.0	47	-	4.8	-
907	-	4.2	44	-	5.2	-
908	-	3.9	41	-	5.4	-
909	-	3.6	49	-	4.2	-
910	-	0.7	73	-	-0.6	-
911	-	1.6	65	-	1.4	-
912	-	2.5	77	-	-1.2	-
913	-	2.7	98	-	-5.4	-
914	-	1.4	5	-	13.2	-
915	-	2.9	12	-	11.0	-
916	-	5.3	48	-	4.2	-

TABLE IX. TAIL ROTOR LOADS AND DISPLACEMENT DATA

Model UH-1B 1092 G.R. 32A Date 18 November 1966
Ship AF64-13968 Tail Rotor Configuration Number 2

CTR NO.	TEST CONDITION (Units Explained in Fig. 13 & Table III.)	MAIN ROTOR RPM	V_{CAL} (KN.)	T/R YOKE CHORD AT 2.1	
				MEAN	OSC.
594	Engine Start to Flt. Idle	220	0.0	-1156	655
595	Flt. Idle to 6750 N2	329	0.0	-1368	905
596	Stabilized RPM	324	0.0	-1734	462
597	Pedal Kicks Slow	324	0.0	-2138	1098
598	Engine Shutdown	-	0.0	-1560	1213

CTR NO.	T/R YOKE BEAM AT 2.1		T/R BLADE CHORD AT 11.0		T/R BLADE BEAM AT 11.0	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
594	-3136	1709	-1360	732	-3428	2303
595	-3927	663	-627	418	-3700	794
596	-3978	561	-732	418	-3678	635
597	-3315	765	-1360	1046	-4177	953
598	-1581	2244	-889	889	-3269	1543

CTR NO.	T/R BLADE CHORD AT 21.5		T/R BLADE BEAM AT 21.5		T/R PITCH LINK RED	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
594	2390	444	-1792	746	-41	22
595	2028	528	-1792	590	-40	16
596	2001	277	-1770	501	-39	22
597	1528	750	-2026	735	-45	36
598	2167	555	-1759	890	-38	14

TABLE IX. - Continued

CTR NO.	T/R SHAFT TORQUE		T/R SHAFT PERP. BEND		T/R SHAFT PARA. BEND	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
594	-2643	1211	-7667	443	-6240	751
595	-2790	357	-7762	411	-5848	816
596	-2918	312	-7793	570	-5782	686
597	-2092	550	-7730	697	-5586	751
598	-2808	1083	-7667	633	-5717	816

CTR NO.	T/R HUB FLAPPING		PEDAL POSITION		T/R BLADE ANGLE	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
594	-	1.7	49	-	5.7	-
595	-	1.9	50	-	5.1	-
596	-	1.7	50	-	5.2	-
597	-	1.7	66	-	0.9	-
598	-	2.7	52	-	5.3	-

TABLE X. TAIL ROTOR LOADS AND DISPLACEMENT DATA

Model UH-1B 1092 Flt. 172A Date 22 November 1966
Ship AF64-13968 Tail Rotor Configuration Number 2

CTR NO.	TEST CONDITION (Units Explained in Fig. 13 & Table III.)	MAIN ROTOR RPM	V_{CAL} (KN.)	T/R YOKE CHORD - AT 2.1	
				MEAN	OSC.
681	Hover	304	0.0	76	667
683	Hover	314	0.0	171	648
685	Hover	319	0.0	248	648
687	Hover	324	0.0	209	648
693	Hover Left Turn	319	0.0	858	1374
694	Hover Right Turn	319	0.0	248	2633
695	Sideward Flight-Left	319	30.0	839	1355
696	Sideward Flight-Right	319	30.0	1984	1316
712	Level Flight	319	62.5	-381	515
713	Level Flight	319	71.5	-534	591
714	Level Flight	319	81.0	-343	706
715	Level Flight	319	90.5	-305	858
716	Level Flight	319	100.0	-248	1183
717	Level Flight	319	110.0	-114	1049
718	Level Flight	319	121.0	-152	1011
719	Level Flight	324	62.5	-400	648
728	Level Flight Into Lt. Turn	319	100.0	-76	1164
729	Level Flight Into Rt. Turn	319	110.5	76	1240
732	Full Power Climb	319	71.5	-19	801
733	Autorotation	319	81.0	-1030	744
734	Autorotation Into Lt. Turn	319	81.0	-839	858
735	Autorotation Into Rt. Turn	319	81.0	-935	801

TABLE X. - Continued

CTR NO.	T/R YOKE BEAM AT 2.1		T/R BLADE CHORD AT 11.0		T/R BLADE BEAM AT 11.0	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
681	-445	657	523	680	444	444
683	-495	708	942	680	577	444
685	-799	708	994	628	599	643
687	-976	632	785	628	532	577
693	313	1112	889	1465	222	710
694	-1026	1138	785	2302	-288	1221
695	692	1188	1046	1413	111	954
696	1325	1466	1517	1256	1731	1110
712	-1203	1213	732	523	-399	532
713	-1052	1264	628	628	-355	710
714	-748	1062	628	418	-133	621
715	-242	859	628	785	-88	888
716	-91	1517	732	889	333	1043
717	136	1694	837	732	377	1221
718	465	2225	732	942	510	1576
719	-1532	885	785	523	-444	577
728	136	1896	732	1099	177	1332
729	60	1770	889	942	310	1287
732	920	1062	1151	575	710	754
733	-3682	1163	157	837	-1221	599
734	-4592	1011	314	785	-1776	666
735	-5022	1441	52	1056	-1731	754

TABLE X. - Continued

CTR NO.	T/R BLADE CHORD AT 21.5		T/R BLADE BEAM AT 21.5		T/R PITCH LINK RED	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
681	-278	361	174	424	-84	45
683	-222	473	119	414	-92	47
685	-111	417	152	424	-81	42
687	-278	417	130	490	-90	33
693	-167	974	43	490	-115	126
694	-222	1586	10	871	-33	135
695	-83	890	-87	751	-169	90
696	194	890	838	697	-236	118
712	278	348	-228	348	53	36
713	-250	389	-261	403	31	42
714	-501	668	-163	414	47	42
715	-612	380	-163	457	39	67
716	-334	528	163	631	36	109
717	222	445	21	817	22	124
718	-278	695	185	1067	25	149
719	-222	361	-283	337	45	39
728	-306	779	-261	926	-47	126
729	-334	640	-98	806	42	109
732	-55	473	196	403	-14	87
733	-835	528	-468	370	188	36
734	-751	501	-784	403	93	25
735	-946	584	-806	468	121	25

TABLE X. - Continued

CTR NO.	T/R SHAFT TORQUE		T/R SHAFT PERP. BEND		T/R SHAFT PARA. BEND	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
681	2015	503	-253	443	-33	864
683	2183	335	-158	475	33	864
685	2165	354	-95	538	99	930
687	2258	335	-126	507	132	831
693	3266	933	-63	1330	-99	1196
694	1735	559	31	1299	0	1495
695	3434	1101	-0	1457	-0	1296
696	5039	914	-221	1299	-0	1695
712	951	298	-31	475	365	930
713	933	242	31	538	232	930
714	951	223	31	475	232	930
715	951	223	-63	570	132	1030
716	1026	447	-221	665	265	963
717	1119	261	-158	728	99	1063
718	1063	373	-221	982	265	1362
719	951	186	-158	475	199	1030
728	1287	335	-190	950	332	1429
729	765	410	-31	602	265	1096
732	1549	298	95	728	432	1263
733	559	466	-95	538	199	764
734	559	391	-126	570	166	864
735	503	298	-190	633	232	930

TABLE X. - Continued

CTR NO.	T/R HUB FLAPPING		PEDAL POSITION		T/R BLADE ANGLE	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
681	-	1.7	31	-	9	-
683	-	1.6	29	-	10	-
685	-	1.2	32	-	9	-
687	-	1.2	32	-	9	-
693	-	3.2	23	-	12	-
694	-	7.1	34	-	9	-
695	-	2.1	16	-	13	-
696	-	2.4	6	-	16	-
712	-	1.8	54	-	4.5	-
713	-	2.3	53	-	4.5	-
714	-	2.9	55	-	4.3	-
715	-	3.5	51	-	4.8	-
716	-	4.1	49	-	5.8	-
717	-	4.9	46	-	6.3	-
718	-	5.4	45	-	6.5	-
719	-	1.7	51	-	4.4	-
728	-	5.3	41	-	7.3	-
729	-	4.3	50	-	5.6	-
732	-	2.6	39	-	7.9	-
733	-	1.2	74	-	-0.4	-
734	-	1.1	71	-	0.5	-
735	-	1.6	73	-	0.0	-

TABLE XI. TAIL ROTOR LOADS AND DISPLACEMENT DATA

Model UH-1B 1092 Flt. 173A Date 28 November 1966
Ship AF64-13968 Tail Rotor Configuration Number 2

CTR NO.	TEST CONDITION (Units Explained in Fig. 13 & Table III.)	MAIN ROTOR RPM	V_{CAL} (KN.)	T/R YOKE CHORD AT 2.1	
				MEAN	OSC.
790	Hover	304	0.0	492	644
791	Hovering Left Turn	314	0.0	1553	1440
792	Hovering Right Turn	324	0.0	663	2671
793	Sideward Flight Left	319	30.0	1137	871
794	Sideward Flight Right	319	30.0	568	1819
795	Acceleration 0 to 40	324	43.0	530	1326
797	Stabilized Level Flight	324	100.0	-265	909
798	Stabilized Level Flight	324	110.0	-265	985
799	Stabilized Level Flight	324	120.5	-246	928
800	Stabilized Level Flight	314	100.0	-113	935
801	Stabilized Level Flight	314	110.0	-246	966
802	Stabilized Level Flight	314	120.5	-322	852
803	Deceleration 40 to 0	324	0.0	-379	1212

CTR NO.	T/R YOKE BEAM AT 2.1		T/R BLADE CHORD AT 11.0		T/R BLADE BEAM AT 11.0	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
790	-100	601	1353	416	763	493
791	876	826	1717	1301	1638	740
792	-1553	1102	1405	2446	-22	1055
793	300	1002	1457	832	1100	1010
794	-50	1352	1197	1613	1122	1436
795	125	776	1509	884	853	898
797	-1753	1152	884	780	202	1010
798	-1653	1452	936	832	89	1077
799	-1728	1377	988	884	179	1122
800	-1152	1052	832	936	157	875
801	-1252	1352	936	832	179	987
802	-1302	1402	832	728	179	1122
803	-2029	1177	1197	988	-628	1212

TABLE XI. - Continued

CTR NO.	T/R BLADE CHORD AT 21.5		T/R BLADE BEAM AT 21.5		T/R PITCH LINK RED	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
790	-0	336	196	479	-101	33
791	-56	1009	610	588	-81	104
792	-168	1683	283	588	-98	132
793	-140	813	599	708	-132	93
794	-448	1178	283	828	-138	[illegible]
795	-168	617	119	664	-19	64
797	-280	504	-196	741	39	124
798	-336	673	-76	773	8	93
799	-280	504	-87	959	31	98
800	-280	673	10	643	81	81
801	-308	645	-32	730	36	121
802	-336	448	-65	763	42	98
803	-364	701	-261	915	-104	64

CTR NO.	T/R SHAFT TORQUE		T/R SHAFT PERP. BEND		T/R SHAFT PARA. BEND	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
790	2935	430	-31	407	198	1057
791	5516	355	62	1254	-165	892
792	2898	841	-125	1128	132	1784
793	3721	804	344	1975	198	991
794	4805	654	156	1222	363	1487
795	3197	467	31	532	297	958
797	1103	317	-0	752	99	1090
798	1215	355	31	846	-33	1024
799	1047	374	188	1191	-132	991
800	710	299	-94	658	-0	661
801	953	280	-125	877	33	826
802	1009	336	31	783	-165	826
803	2337	542	-62	627	99	1090

TABLE XI. - Continued

CTR NO.	T/R HUB FLAPPING		PEDAL POSITION		T/R BLADE ANGLE	
	MEAN	OSC.	MEAN	OSC.	MEAN	OSC.
790	-	1.1	27	-	11	-
791	-	2.3	22	-	14	-
792	-	7.1	23	-	11	-
793	-	2.6	21	-	13	-
794	-	2.0	14	-	14	-
795	-	1.9	24	-	11	-
797	-	3.5	50	-	5	-
798	-	4.1	49	-	5	-
799	-	4.4	50	-	5	-
800	-	3.9	55	-	4	-
801	-	4.1	51	-	5	-
802	-	5.0	50	-	5	-
803	-	2.7	31	-	10	-

APPENDIX II
ELASTOMERIC-BEARING TAIL ROTOR DYNAMIC CALCULATIONS

BONDED ELASTOMERIC RADIAL BEARINGS (Flapping) - Thrust Variations - Oscillatory thrust variations of the two-bladed tail rotor are of relatively low magnitude at a frequency of two-per-rev or approximately 52 cps. The natural frequency of the rotor mass acting on the two trunnion bearings is determined as follows:

Mass of rotor $M_R = \frac{29.5 \text{ lb.}}{386 \text{ in./sec}^2} = .0765 \frac{\text{lb.-sec}^2}{\text{in.}}$

Bearing radial spring rate $K_T = 2 \times \frac{500}{.00174} = 574{,}000 \frac{\text{lb.}}{\text{in.}}$

Natural frequency in the thrust direction

$$\omega_T = \sqrt{\frac{K_T}{M_R}} = \sqrt{\frac{574{,}000}{.0765}} = (7.5 \times 10^6)^{1/2} =$$

$$2.74 \times 10^3 \text{ rad./sec.} = 437 \text{ cps}$$

The amplification factor is determined from the ratio of the excitation frequency to the natural frequency:

$$A_F = \frac{1}{\sqrt{\left[1 - \left(\frac{\omega}{\omega_N}\right)^2\right]^2 + \left[2\,\zeta\,\frac{\omega}{\omega_N}\right]^2}}$$

For the rotor, assuming the damping factor (ζ) = .05,

$$A_F = \frac{1}{\sqrt{\left[1 - \left(\frac{52}{437}\right)^2\right] + \left[2\,(.05)\,\left(\frac{52}{437}\right)\right]^2}} = 1.029$$

The effect of the trunnion bearing spring rate on motions in the thrust direction is negligible.

BONDED ELASTOMERIC RADIAL BEARINGS (Flapping) - Drive System Oscillations - Drive system torsional oscillations are generally related to the first two natural frequencies of the drive train: 3.3 and 8.3 cps. For large flapping angles of low magnitude, two-per-rev (52 cps) may occur due to Hooke's joint effects.

The natural frequency (ω_Q) of the rotor on the trunnion bearings is determined as follows:

Inertia of rotor $\quad I_R = 30.54$ in.-lb.-sec.^{2}

Bearing spring rate $\quad K_Q = 2K_R \ell^2 = (2)(287{,}000)(1.375)$
$= 789{,}250$ in.-lb./rad.

where ℓ is the distance from the shaft axis to the bearing, and K_R is the spring rate of the rotor flap bearings in compression, radially.

$$\omega_Q = \sqrt{\frac{K_Q}{I_R}} = \sqrt{\frac{789{,}250}{30.54}} = (25{,}843)^{1/2}$$

$$\omega_Q = 160.7 \text{ rad./sec.} = 25.57 \text{ cps}$$

Since this frequency is well removed from the predominant excitation sources, no dynamic effects are anticipated.

Note that this spring rate is much greater than that of the tail rotor mast of 259,000 in.-lb./rad.

BONDED ELASTOMERIC RADIAL BEARINGS (Flapping) - Flapping Response - The flapping natural frequency of a teetering rotor with no pitch-flap coupling (δ_3) and no hub spring is one-per-rev, or

$$\omega_F = \sqrt{\omega^2_{ST} + \Omega^2}$$

where ω_{ST} = natural static frequency = $\sqrt{\frac{K_F}{I_F}}$

and where $K_F = 2\ K_T$ = the torsional spring rate of two trunnion bearings.

$$K_F = 2(6.225 \frac{\text{in.-lb.}}{\text{deg.}}) = 12.45 \frac{\text{in.-lb.}}{\text{deg.}} \times \frac{57.3 \text{ deg.}}{\text{rad.}} = 714 \frac{\text{in.-lb.}}{\text{rad.}}$$

and

$$I_F = 30.54 \text{ in.-lb.-sec.}^2$$

$$\omega_{ST} = \sqrt{\frac{715}{30.54}} = 4.84 \text{ rad./sec.}$$

$$\omega_F = \sqrt{(4.84)^2 + (163.4)^2} = \sqrt{23.4 + 26699} = \sqrt{26722.4}$$

$$\frac{\omega_F}{\Omega} = \frac{163.505}{163.5} \cong 1.0/\text{rev.}$$

This should not produce any significant dynamic loading as a result of the change in frequency.

Another effect is produced, however, by the damping of the bearings. From vibration tests, the damping of the flapping mode was determined to be 18 percent of critical.

A shift in phase of the azimuth for maximum flapping, as compared to a rotor with no δ_3 and no flapping moment spring, may, in turn, affect the aerodynamic loading and thus the blade bending moments.

BONDED ELASTOMERIC THRUST BEARINGS - Pitch Change with Flapping- The pitch links of the rotor are located on the trunnion of the flapping axis. Thus no mechanical δ_3 is introduced except through the geometric changes due to flapping. For maximum flapping of ±8 degrees, the maximum pitch change is approximately 0.5 degree, occurring at the flapping frequency.

Restraint to pitch change is produced by the thrust bearing action in torsion under 15,800-pound centrifugal loading. The magnitude of this restraint may be determined as follows:

The feathering spring rate:

$$K_\theta = \frac{255 \text{ in.-lb.}}{12 \text{ deg.}} = 21.2 \frac{\text{in.-lb.}}{\text{deg.}}$$

The pitch change moment:

$$M_P = K_\theta{}^\theta = 21.2 \text{ in.-lb./deg.} \times 12 \text{ deg.} = 22 \text{ in.-lb.}$$

The pitch link loads F_P produced will be:

$$F_P = \frac{M_P}{\ell_P} = \frac{255}{2.375} = 107 \text{ lb. at 26 cps}$$

where ℓ_P is the distance from the pitch link to the feathering axis.

BONDED ELASTOMERIC THRUST BEARINGS - Control Inputs - Rudder pedal inputs will effect a change in pitch and will be reacted by the thrust bearing spring rate, as in the case of flapping. However, for maximum rate inputs, the pitch link loads will be increased due to the damping for the thrust bearings, for the Teflon bearings, and for large flapping angels by the damping in the trunnion bearings (due to geometric changes with flapping).

TEFLON JOURNAL BEARINGS

Loading: The Teflon bearings are loaded as follows:

Source	Type of Loading	Direction
Driving Torque	Steady	Radial (Inplane)
Drive System and Blade Chord Bending	Oscillatory	Radial (Inplane)
Thrust	Steady	Radial (Out of Plane)
Blade Beam Bending	Oscillatory	Radial (Out of Plane)

Tests of Teflon bearings oscillating under load have shown that the coefficient of friction decreases with load and increases with velocity. A resultant steady load of approximately 700 pounds is produced by a thrust of 500 pounds and a driving torque of 3500 in.-lb.

A surface velocity ($V = R\theta\omega$) of 14.2 fpm is produced by a pitching oscillation of 0.5 degree at 26 cps.

The coefficient of friction is estimated for 700 psi and 14.2 ft./min. as $\mu = .15$.

The friction force $f = \mu N = (.15)(700) = 150$ lb.

The friction torque $q = fr = (105)(1) = 105$ in.-lb.

The pitch link load is $F_p = \frac{q}{l} = \frac{105}{2.375} = \pm 44.25$ lb. at 1/rev.

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13. ABSTRACT The results of a program conducted to investigate elastomeric bearings is presented. Phase I of this program involved fabrication and bench testing of the bonded-type elastomeric bearings. Bench tests showed that two thrust bearings of this type would be required per grip to carry the blade centrifugal force. An experimental tail rotor assembly utilizing the radial and thrust bearings was designed and fabricated. Whirl and shake tests were conducted to determine rotor natural frequencies, dynamic spring rates, and bearing durability. Phase II of the program consisted of flight testing of the bonded-thin-layer-type bearing tail rotor. During the latter part of the program, a molded-type elastomeric thrust and radial bearings were designed and fabricated. The contract was modified to include flight testing of the molded-type thrust bearing which required only one thrust bearing per blade grip to carry the centrifugal force and provide the pitch change motions. Results from both configurations compared favorably with standard UH-1 tail rotor data except for higher Sta. 21.6 blade oscillatory loads. The higher oscillatory loads, which would limit the allowable blade useful life, were attributed to tail rotor natural frequencies occurring near the helicopter operating three-per-rev. resonance. The second configuration is more attractive from a hardware, assembly, and flight operation characteristics standpoint. The molded-type elastomeric bearings were found to be marginally satisfactory for this application.

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	ROLE	WT	ROLE	WT	ROLE	WT
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